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July, 1941

Heating · Piping and Air Conditioning

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L — Although the U. S. oduces roughly 65 per t of the world's petro- m supply, transfer to tain of tankers serving the antic Seaboard has caused despread speculation on fuel oil shortage there. lman Steiner reviews the uation in this month's AC, tells why he believes use of fuel oil should expanded. See p. 417

Page 439
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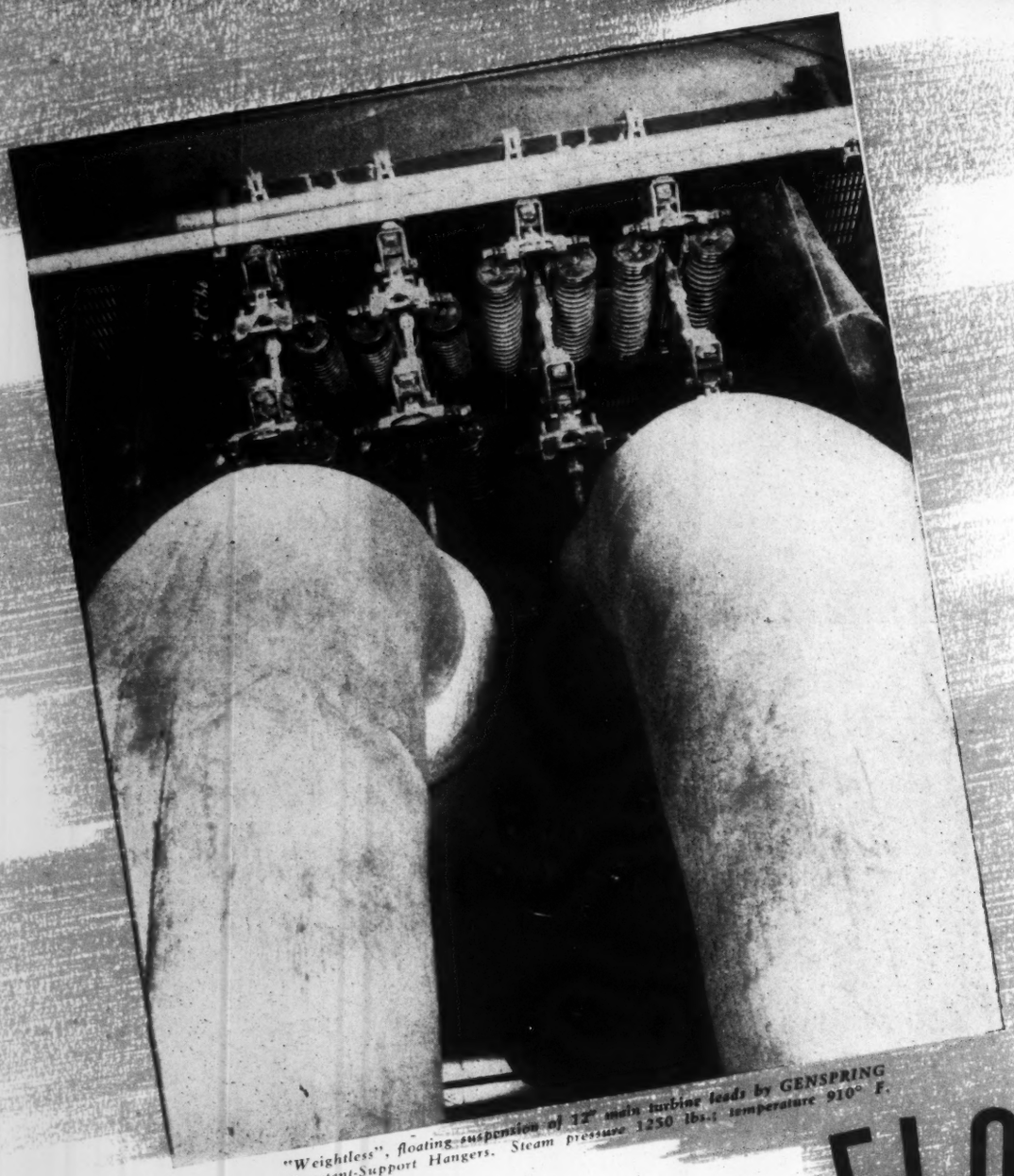

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WHENEVER PIPING IS INVOLVED

The Editor's Pages

AMERICAN INDUSTRY TURNS TO AIR CONDITIONING FOR HELP IN SOLVING PRODUCTION PROBLEMS

A quarter of a century ago—while America was struggling against time to equip a mighty army and navy—the "infant" air conditioning industry stepped into the national emergency for the first time to solve several production problems in ammunition, fuse loading and armament plants.

Today, the air conditioning industry has taken its place in the "front line trenches" of America's gigantic defense program and is helping to step up production and turn out better airplanes, tanks, ammunition and bombs, iron and steel, engines, machine tools, range finders and other instruments, as well as food, electrical products and many other items indirectly required in national defense.

When the Wright Aeronautical Co., Lockland, Ohio, contracted last month for five centrifugal refrigerating machines of 1200 ton capacity each for its new completely air conditioned, 2,000,000 sq ft windowless plant, it was just one indication of the extent to which defense industry in America is turning to air conditioning to help solve its high speed production problems, Dr. Willis H. Carrier, chairman of the board of Carrier Corp., pointed out.

"In order to maintain accuracy and uniformity of quantity production, it is vitally necessary to control the physical properties of materials and the chemical or biological reactions of processes that enter into manufacture," Dr. Carrier said.

"Expansion and contraction of metals under variation of temperature often exceeds the tolerances permitted in the manufacture of airplanes, tanks, engines and other machines."

During machining operations in machine shops, air conditioning is often required to maintain uniform temperatures. Dr. Carrier stated that even slight temperature changes in master gage rooms where gages accurate to millionths of an inch are lapped and inspected, would cause the metals to change dimensions. Furthermore, humidity must be carefully controlled to guard against perspiration from the workers' hands marring the extremely high finishes of the gages.

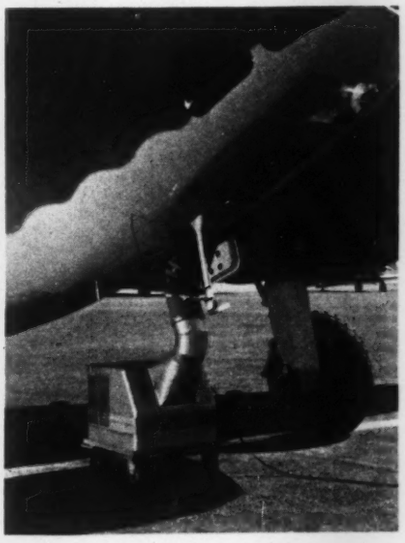
Another important use of air conditioning is in the testing of airplane engines, according to Dr. Carrier. Airplane engines must perform under a wide range of conditions involving severe changes in temperature, humidity and air pressure. Many engine test rooms now employ air conditioning to vary the conditions of air supply to the carburetor.

Still other uses have been found in the manufacture of instruments such as chronometers, range finders, lenses and optical goods.

EVAPORATIVE COOLING FOR BOMBERS

Evaporative air coolers are now being used to reduce the temperature in Lockheed bombers while interior construction is completed and instruments are installed. As the big twin engined bombers reach final assembly, they are rolled out of the factory onto the apron at Lockheed field, Burbank, Calif. Under the California sun, temperature in the metal fuselages reached a point beyond comfort, so air coolers are connected to the flare chute. The coolers are a standard window model; this type has a water reservoir and recirculating pump, so a water connection is unnecessary

Photo courtesy Utility Fan Corp.



STOKER MAKERS PLEDGE DEFENSE SUPPORT

Stoker manufacturers from all parts of the United States gathered at White Sulphur Springs, W. Va., for the 24th annual meeting of the Stoker Manufacturers Association and pledged full support to the defense and emergency program. They agreed to devote not only as much plant capacity, manpower, and material as possible for defense production, but to increase stoker production to almost any extent required for heat, steam and power, to meet the rapidly expanding demand from industry and government.

With stoker sales way up for the first five months of 1941 as compared to the corresponding period of 1940, members of the SMA expect a record-breaking year, unless priority and other defense requirements or emergency conditions interfere. While the attention of the stoker executives centered on the effect of defense on the stoker business, it was the unanimous feeling that the stoker industry has a brilliant future.

STOKER SALES GO UP

Seventy-eight stoker manufacturers furnishing data to the United States bureau of the census, of the 101 manufacturers on the list, reported factory sales in April this year totaling 10,139 units of all sizes and types. This is an increase of over 53 per cent compared with the same month in 1940.

Sales for the first four months of this year totaled 31,140 of all sizes and types of stokers, compared with 18,995 in the corresponding months of 1940.

A Worthwhile Opportunity

Engineers in charge of the operation and maintenance of large heating, piping and air conditioning systems in plants and buildings—and those responsible for the design and installation of these services — have a particularly worthwhile opportunity in these days of defense activity and higher taxes, which (in one way or another) place a premium on efficiency. Efficient operation of the mechanical equipment of a building always pays dividends. But with rising costs of doing business and higher taxes to come, savings of this kind are even more important.

A case in point is given in the article by A. J. Rummel in this month's issue of HEATING, PIPING AND AIR CONDITIONING. Mr. Rummel recently completed an extensive study of the operation of large air conditioning systems in the Southwest. In a typical case analyzed in his article, he found that by better operation of the system — not running the compressors when the outside temperature conditions allowed—that in one month over $160 worth of electricity could have been saved. Analysis of many other installations would undoubtedly reveal the same or similar opportunities for cutting operating costs—or, to put it in another way, increasing profits.

Operating engineers also have another responsibility—conservation for defense. Kalman Steiner, discussing the fuel oil situation in his article this month, indicates the necessity of saving fuel by more efficient combustion and utilization—not only oil, but other fuels as well. Mr. Steiner is preparing two other articles which will recapitulate recommendations he has given in the past for reducing fuel consumption for heating.

START WORK ON CONVECTOR CODE

Formation of a joint committee consisting of two representatives of the Institute of Boiler and Radiator Manufacturers and two members of the Convector Manufacturers Association was effected at a recent meeting of these two groups.

Function of the committee will be to develop a code for testing cast iron and copper convectors, supplemented by a code for the rating of such convectors, and to incorporate the details of these codes in a commercial standard in which an effort also will be made to standardize sizes and enclosures.

It has been recognized for quite some time that a reliable and uniform method of determining boiler, radiator and convector ratings was desirable, and formation of this committee provides a way to solve various problems of this character. Perhaps the most immediate result of this effort will be to obtain a uniform basis for comparison of radiators and convectors by various federal bureaus.

McAWARD NEW NACA SECRETARY

A. C. Buensod, treasurer of the National Air Conditioning Association, announced last month the resignation of Arthur F. Callahan as secretary of the association, and the appointment of James C. McAward as secretary. The offices of the association remain at 220 E. 42nd St., New York City.

"ECONOMIC COMPULSION" FORCES CONDITIONING

Air conditioning is installed primarily because of "economic compulsion," Realto E. Cherne, engineer, Carrier Corp., declared at the second annual air conditioning institute at Massachusetts Institute of Technology last month. Discussing the economics of air conditioning, Mr. Cherne cited the necessity for manufacturers and merchants to guard against losing customers to competitors as an underlying reason for many air conditioning installations.

Information obtained in recent years on actual operating costs on many varied types of air conditioning installations is taking some of the guess work out of computing these costs, Mr. Cherne told the institute members. Up to about 10 or 12 years ago, very few air conditioning systems were equipped with sufficient meters to allow keeping accurate records.

"It is interesting to learn that these costs can be related to the installed cost of the system and the average annual cost of power, water, steam, maintenance, repairs and labor is about 7 per cent of the first cost," he said. "This percentage varies from about 5 per cent for high priced systems to about 10 per cent for very inexpensive installations."

Sales of central plant air conditioning systems in Chicago continued their strong upward trend in May, figures released last month by Commonwealth Edison Co. reveal. Contracts were reported for 79 installations, a new all-time high for May.

Combined capacity of the systems sold was 1161 hp, as against a total of 677 hp for the 63 installations contracted for in May, 1940. Restaurants again led the list.

"IT'S MODERN DESIGN"— The conical shaped old Sibley tent stove, about which legends of '17 cling as thickly as pine wood soot, is to be replaced by a new type, according to the war department. The new stove, developed at the Jeffersonville quartermaster depot, is collapsible like a picnic drinking cup, and is composed of four detachable parts. Cylindrical in shape, it has a base diameter of 18 in., a 4 in. chimney opening at the top, and it weighs approximately 45 lb.

Besides this improvement in design, which facilitates storage and shipment, the new stove has a grate and can burn any hard fuel. The old Sibley was intended as a wood burner as it had no grate and therefore could not burn hard fuel successfully, though many a soldier kept trying on cold winter nights.

The parts of the new stove are completely interchangeable with the old Sibley, thus making it possible by the addition of the grate to burn hard fuels in the old type with which most of the present camps have been equipped. Another advantage of the new stove is its flat top, upon which water can be heated for washing and shaving, and even a little impromptu cooking performed.

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IN THIS ISSUE

Volume 13 July, 1941 Number 7

The Editor's Pages (current comment and news).....front section 39

Keep 'Em Flying!—Army Air Corps' New Slogan Involves Carefully Planned Engine Testing Facilities, by F. W. Swan, Jr..... 405

District Heating Men Meet (highlights of NDHA convention), by G. D. Winans 410

Unused Ice Plant Capacity Serves Albeth Skating Rink (the piping described), by F. S. Clark..... 412

How Air Conditioning Aids Defense (manufacture of precision instruments), by Edward T. Murphy..... 414

It's a Job (to find space for air conditioning equipment in existing buildings—how it was done in a bank), by W. E. Lowell..... 415

Conditioning Bombproof Underground Spaces (brief abstract)..... 416

Expand the Use of Fuel Oil (review of the fuel oil situation), by Kalman Steiner 417

Check Piping for Savings (excerpt from power conference paper), by R. S. Hawley..... 419

The Water-Air Chart (its use as a psychrometric chart), by William Goodman 420

Air Conditions for Efficiency of Draftsmen..... 422

Gasketed Joints for High Pressure, High Temperature Piping Service (including recommendations based on experience), by Arthur McCutchan 423

Increased Taxes and Selling Costs Make Efficient Operation in Plants and Buildings Much More Important (how to operate air conditioning plants to cut power costs), by A. J. Rummel..... 428

Construction Details for Radiant Heating and Cooling (test installation at Northwestern Technological Institute), by T. Napier Adlam..... 431

Production for National Defense—Ventilation of Plating Tanks, by Allen D. Brandt 434

"Open for Discussion" (communications to the editor)..... 438

Equipment Developments .....back section 116

Useful Information .....back section 131

Book Reviews .....back section 134

Coming Events .....back section 139

Trade Literature .....back section 140

Index to Advertisers.....back section 146

Journal Section of the American Society of Heating and Ventilating Engineers

The Interaction Constant for Moist Air, by John A. Goff and A. C. Bates 442

Interpretation of Summer Weather Temperature Data for Design, by J. Donald Kroeker and Harold V. Soballe..... 450

Local Cooling of Workers in Hot Industry, by F. C. Houghten, M. B. Ferderber and Carl Gutberlet..... 462

Bulletin Section 466

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Keep 'Em Flying!

ARMY AIR CORPS' NEW SLOGAN INVOLVES CAREFULLY PLANNED ENGINE TESTING FACILITIES

By F. W. Swan, Jr.

AIR cooled airplane motors depend upon the speed of flight to draw sufficient air over the cylinder heads to prevent overheating. In at least three instances, however, it is necessary to run them in the testing laboratory or factory before they are mounted on a plane:

(1) A great deal of fundamental research must be done on a single cylinder. This is exhaustive and accurate work and often requires the maintenance of the exact temperature, humidity and air pressure conditions which will be encountered in actual flight.

(2) After it has been assembled, each new type of plane motor must undergo a dynamometer test.

(3) On torque stands, where they are run in or given endurance tests, engines have heretofore been loaded by a club propeller which in turn provided the cooling air. Recently engines have been increased in power with very little change in overall diameter. This means that with present design a propeller large enough to load the motor will pass insufficient air near the hub to cool it. Thus, external cooling fans are required to supplement the club propellers.

(4) A rather important instance which calls for external cooling fans occurs during tune up periods. This may be at any time during the life of a motor when some adjustment or inspection is undertaken, necessitating operation for an appreciable time either in the plane on the ground or in the repair shop.

KEEP 'EM FLYING!—is the new slogan adopted by the army air corps; one of the most important considerations in living up to it is complete facilities for testing single cylinders or complete airplane engines in the laboratory or factory before they are mounted on planes, and torque stands for running in completed motors and for repair work. Mr. Swan, research engineer, Buffalo Forge Co., presents a complete discussion here of methods of supplying cooling air for these purposes, and describes in some detail a recent installation, which is one of the largest ever built. . . . Developments in design of air cooled airplane engines have of course been reflected in the necessary test arrangements, and these are reviewed by the author. Flexibility of the test setup is important, as a great variety of motors may now be built by a single manufacturer. Mass production of planes and plane motors has created a demand for cooling fans in many factories, army fields and in repair shops

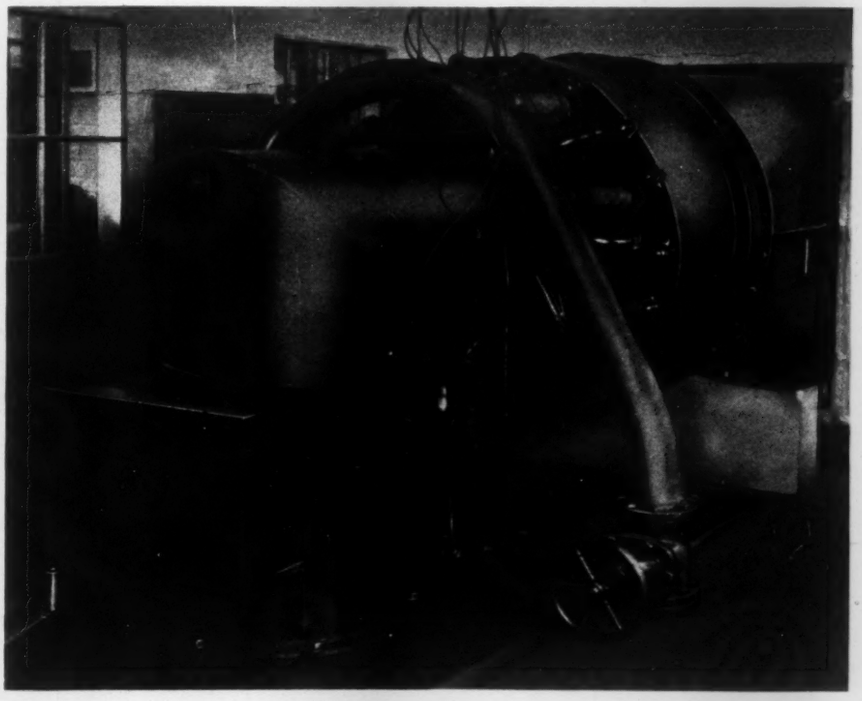


Fig. 1—Early engine cooling installation

Single Cylinder Cooling

As mentioned, one of the two clearly defined testing operations is the setup and observation of a single cylinder at a time. This is a fundamental test which may last for months before a complete motor is assembled or even entirely designed. The second case is the power test of a complete engine. The latter requires considerably more space and more air handling equipment. Both methods are widely used since each has its particular advantages and applications.

About 17 years ago one of the first engine cooling test installations was made for a single air cooled cylinder. Since then single

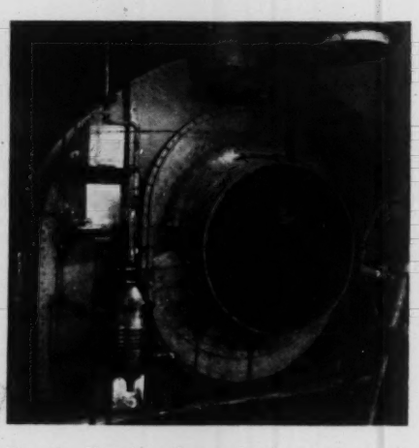


Fig. 2—Hood and nozzle for test stand

cylinder cooling tests have been performed with one of perhaps three different equipment layouts.

The system of Fig. 3a requires the maximum in vision plates and access doors for observers, since the object of the test is entirely enclosed in the ductwork. An investigation of the resistance to air flow will show that the cylinder itself acts as an abrupt change in pipe diameter so that the calculation of fan requirements as to air and total pressure must be based upon this type of flow.

In Fig. 3b, the cylinder is merely an obstruction to the air, as it blocks off a percentage of the nozzle outlet. If the cooling fan has been selected and rated correctly, then when the cylinder is mounted the air velocity will automatically increase to the design conditions.

An additional section of ductwork, Fig. 3c, to pick up the high velocity air after it has passed the engine cylinder is useful since it removes a terrific blast of air from the room in which observers may be standing. This also means less space required for the air discharge opening. Ordinarily, the nozzle and the re-entrant cone will be mounted upon swinging elbows in order that the setting up and servicing of the test cylinder may be accomplished with the greatest facility. A flexible arrangement such as this also lends itself to a variety of test subjects with a minimum of change in the permanent ductwork.

Complete Engine Cooling

In the test of a complete airplane motor many facts and theories may be checked which can only be predicted from observation of a single cylinder.

Since the first airplane motors were mounted entirely in the open, with no shroud or covering over the cylinder heads, it was possible to cool them during tests by merely shooting a blast of air at them from a nozzle. This method required a great deal of air, since the main ductwork had to be extremely large in order to keep the velocity (and, therefore, the resistance) at a reasonable figure.

The measurement of power developed by an airplane engine is of prime importance and requires considerable equipment. The dynamometer stand should be as close as possible to the motor, but an unrestricted air flow is also required. One of the first installations was made as shown in Fig. 4a. The air came from above, went around an abrupt elbow and then into a nozzle which contained an egg crate grid, the purpose of the latter being to provide even distribution of air over the motor. It was often built in an octagonal shape about 48 to 60 in. in diameter. In order that the dynamometer shaft overhang might be as short as possible, the duct elbow was nearly a right angle and airfoil splitters were installed in this elbow to prevent uneven flow at the entrance to the grid.

Needless to say, many other layouts have been designed to mount the dynamometer close to the test motor. Two fans are sometimes operated in parallel, each blowing into a branch of a Y connection as shown in Fig. 4b. This arrangement should only be used with centrifugal fans which have backward or double curved type blades; otherwise, when the volume is dampered off, unbalanced operation and "hunting" may occur, with resulting difficulty in maintaining stable operation and a fixed air velocity.

Airplane designers later added a cover or shroud over the cylinders of air cooled engines. This item, of course, had to be duplicated in the laboratory and the result was as follows: The engine being tested was moved into the end of a carefully designed nozzle which supplied the cooling air. The ductwork thus acted as a shroud. Since the outlet was no longer an open pipe as large as the motor, the cfm of air required for test purposes was greatly reduced. As years went by, the government and private concerns built new and bigger laboratories and multiple setups were introduced, Fig. 4c. In addition, a great variety of motors may now be built by a single manufacturer and this necessitates a flexible arrangement.

Because of the large horsepower rating of most plane motors, several laboratory installations have been made in which the motor tested is connected to an electric generator which in turn supplies current to the fan motor; this conserves power which ordinarily would be wasted.

A Recent Installation

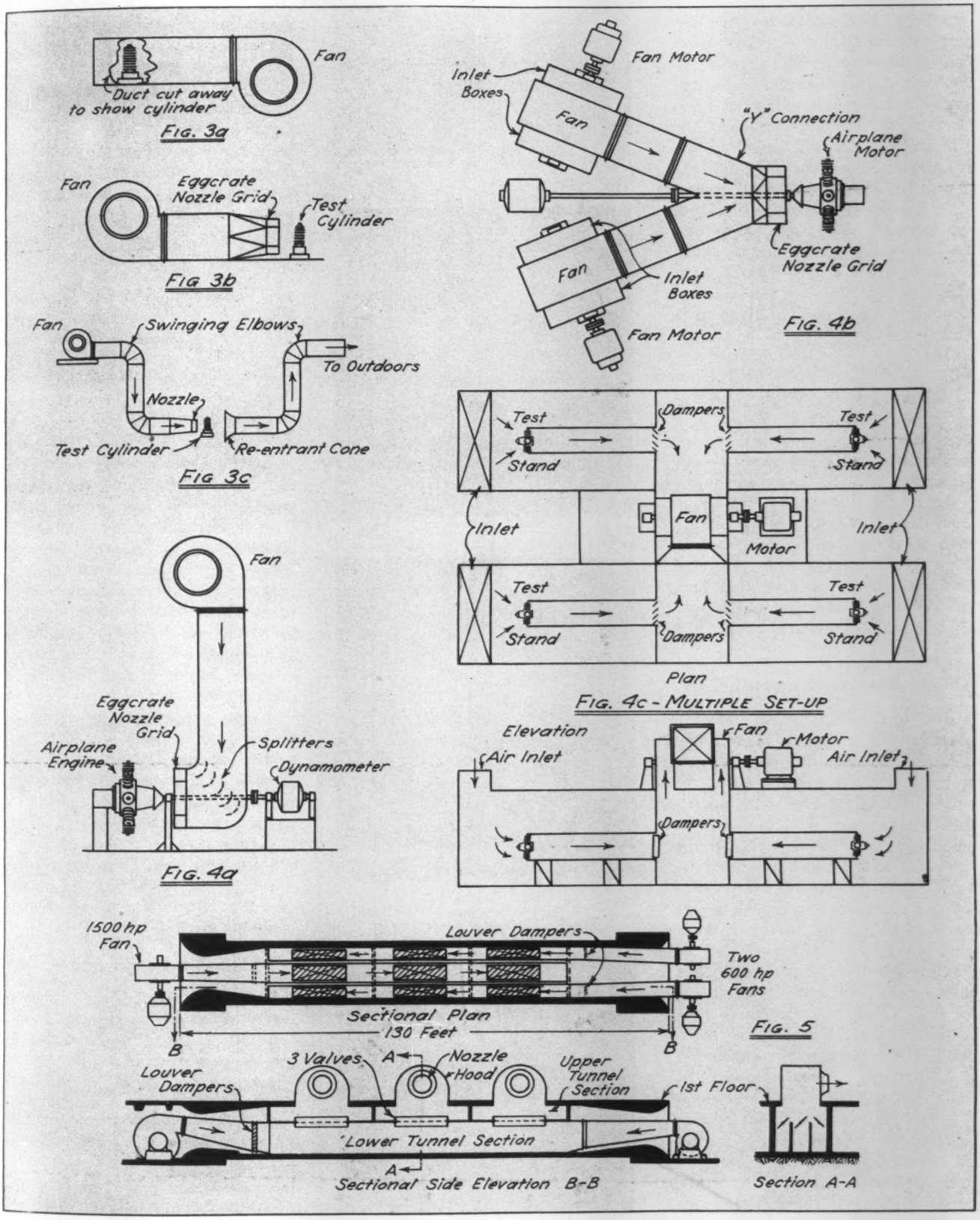
A recent installation—one of the largest ever built—was designed to create an air stream traveling at 366 mph. Some conception of the intensity of an air velocity of this magnitude may be obtained by recalling that the West Indies hurricanes which do so much damage are rarely accompanied by winds in excess of 150 mph. Fig. 5 shows this installation, with three separate test stands mounted over a horizontal square tunnel.

The previous setup consisted of two propeller type fans, each of which was capable of handling 40,000 cfm against 16 in. static pressure when consuming 600 hp. With these units the maximum air velocity that could be created was 180 mph. In addition, the inefficiency of the fans (17 per cent mechanical)

would at times heat up the air handled as much as 60 F before it reached the testing stand.

In order to enlarge this installation, three centrifugal fans of the double curved, multiblade type were purchased. Two of these (72,000 cfm air capacity at 33 in. static pressure) are situated at one end of

Fig. 3—Single cylinder test setups. . . . Fig. 4—Various types of dynamometer stands. . . . Fig. 5—Sectional views of test hoods and supply tunnel



the tunnel, Fig. 6, which tunnel is 12 ft square and about 150 ft long. Two of the fans are driven by the two original 600 hp motors, while the third (handling 100,000 cfm at 65 in. static pressure) requires a 1500 hp driving unit, Fig. 7.

All of the motors are the slip ring type and may be varied from full to half speed in 10 equal steps. In spite of the greatly increased capacity and pressure, the centrifugal fans are of such efficiency that the temperature rise of the air when passing through the 600 hp units is only 16 F with over twice the pressure previously attained. The useful work obtained from these motors is nearly four times as much as in the past. The 1500 hp fan raises the air 35 F but the pressure developed by this fan is approximately 2½ psi.

There are dampers directly beyond the fan outlets. The 12 ft x 12 ft tunnel is divided into an upper and lower section, Fig. 5, and the lower half is partitioned lengthwise so that each fan blows into a separate duct. The upper half is cut transversely into thirds and above each third is a test stand consisting of a hood and nozzle, as shown in Fig. 2. Automatically controlled dampers are so situated that one, two or three of the fans may supply air to any or all of the nozzles at the same time.

Control Rooms

The test hoods are in separate dynamometer rooms, each of which is flanked by a sound insulated control room. In the latter are all of the indicating and recording instruments for engine test data as well as selector control panels (Fig. 9) for the plane motor and for each of the fans. Provided no other station is using a certain fan, complete acquisition of control of that fan may be obtained at any station by merely pushing a button. This action opens the proper tunnel valve between the longitudinal duct in the lower tunnel section and the transverse chamber in the upper tunnel section. At the same time, complete control of fan speed and throttle damper position is assigned to that one station exclusively. When testing is completed at that stand, the pushing of a button stops the fan, closes the tunnel valve and releases the control so it may be acquired by either of the other two stations.

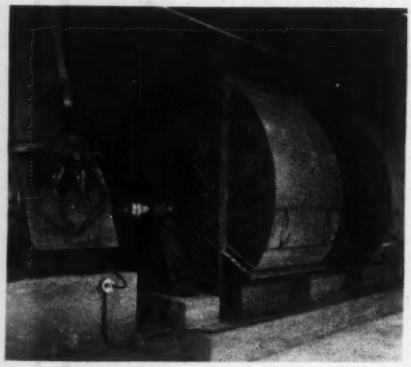


Fig. 6—The two 600 hp fans and motors

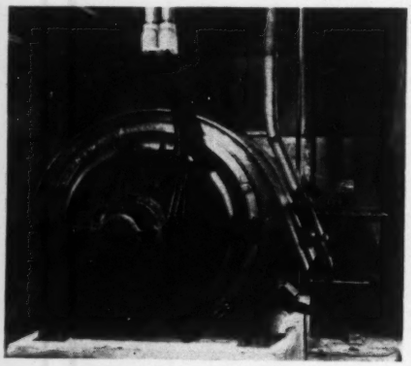


Fig. 7—Fan and motor at north end of tunnel

Fig. 8—Looking down through hood

Fig. 9—The motor control room

The control of cooling air consists of varying the fan from full to half speed in 10 equal steps. Velocities below 50 per cent of maximum are obtained by running at half speed and partially closing the throttle dampers at the fan discharge. Throttling dampers are controlled by a pushbutton station; one opens the dampers slowly and another closes them slowly. An interlock will not allow the fan to run above half speed until the dampers are fully open, thus preventing operation at a higher speed than necessary with a consequent waste of power. An air speed gage, calibrated directly in mph, in the engine control room is connected to a pitot tube at the nozzle. Observing this, the operator may vary the damper position or the fan rpm to meet the cooling requirements.

The open or closed positions of the dampers and valves, the on or off operation of the motor and the availability of control of any of the units are all indicated on selector panels by red and green signal lights. An emergency stop button is provided to trip the primary breakers and close the tunnel valve regardless of the operating speed or damper position. In case of power failure at the fan motor, a glaring light, connected to a separate source of power, is provided in each engine control room.

The motors operate on a primary circuit at 6600 volts while the secondary control circuit is 220. To provide all the automatic and safety features, considerable secondary control equipment is needed. This is placed in the attic. Fig. 10 shows the front of the secondary control panel for the two 600 hp motors and Fig. 11 illustrates the resistor bank for the 1500 hp motor.

Heavy Construction

The entire system may be subjected to any pressure up to 2½ psi. In addition, the air temperature may vary from below zero in winter up to 140 F in summer. There are diaphragm type expansion joints at each fan outlet. The breeching from the fans to the tunnel, the horizontal division sheet in the tunnel and other sheets which have air flowing on one side only

are ¼ in. plate reinforced with 4 in. channels on 3 ft centers. The vertical partition walls which divide the lower part of the tunnel into three ducts may have air flowing on both sides and since exposed bracing members would add considerably to the air resistance—and therefore the power consumption—these walls are double panel construction with sheets welded to both sides of 4 in. channel irons. Further reinforcement is provided by 2 in. pipe stays running between the partitions and from the latter to the concrete tunnel walls.

The throttling dampers at the fan discharges are double plate, louver type, rubber tipped and ball bearing mounted. Alternate blades rotate in opposite directions to give more uniform dampering throughout the range and to minimize the torque. The operating mechanism is a motor driving through double worm and wheel speed reducers, with limit switches at either end of the travel. The blades require 20 sec to move from full open to full closed. The resulting gradual change in air flow gives ample time to stop the dampers at the desired position as indicated by the air speed gage.

Nine valves between the lower and upper tunnel sections are of the rubber seated type, single bladed and center pivoted on 4½ in. ball bearings. When closed, the machined edges of the valves compress a seat of pure live rubber ¾ in. thick which tightly seals the opening against the 2½ psi air pressure. These valves are also motor operated through reduction gears protected by limit switches. These are shown in Fig. 8 and Fig. 12.

The 1500 hp fan rotor is over 7½ ft in diameter and at 1200 rpm has a peripheral speed of nearly 30,000 fpm. To withstand the centrifugal stresses imposed by this speed, the wheel was constructed of alloy steel having a tensile strength of 100,000 psi. The fan wheels were given an accurate dynamic balance and the housings have access doors in the scrolls for cleaning, since at these high peripheral speeds an uneven accumulation of dirt could easily unbalance the entire rotor assembly.

Fig. 10—Secondary control panel for 600 hp motors

Fig. 11—Resistor bank for 1500 hp motor

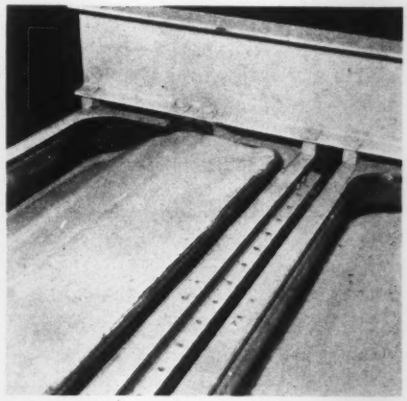
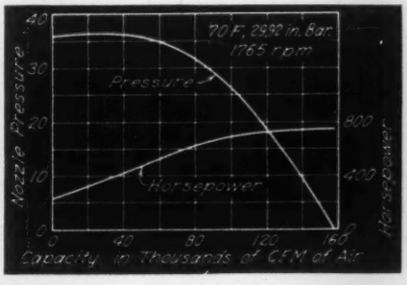


Fig. 12—Close-up of tunnel valve

Fig. 13—Performance curves for one of the 600 hp fans



The arrangement of the system makes an accurate check of air capacity a simple matter. Substitution of various diameter thin plate orifices at the nozzle outlets permits the demonstration of guaranteed fan performance and the determination of the system characteristics.

The performance curves of one of the 600 hp fans is reproduced in Fig. 13. It can be seen that the self-limiting horsepower curve permits operation at any air capacity, at the design speed, without the possibility of imposing more than reasonable overload on the motors. Even if the fans could be operated at a free discharge condition, the motors would never be overloaded more than 25 per cent. System resistance, of course, prevents anything approaching free delivery operation.

Torque Stands

Just as in the case of testing a newly designed airplane motor, club propellers have been widely used on torque stands in order to run in each completed engine. Today the cowlings hamper the cooling effect of this propeller so that additional air is needed.

After all experimental testing has been done it is possible to design a club propeller so that it can be definitely stated that at a certain speed it will load the engine to an accurate and predetermined figure. With the load fixed, the amount of cooling air may be calculated so that it duplicates that encountered in flight. Then, by means of many thermocouples, the temperature during the running in period can be measured at almost any point. This will give an indication of the soundness of a motor, since hot spots or imperfect firing will show up on the instrument board as a too low or too high temperature reading.

Mass production of planes and plane motors has created a large field for cooling fans in many factories, army fields and privately owned repair shops. Often heavy duty centrifugal fans supply air to these torque stands. Recently, however, the axial flow type fan with a limit load horsepower characteristic has come into favor because of its ease of installation and the saving in space and ductwork.

DISTRICT HEATING MEN MEET

and talk about sales, economical heating, steam generation and use, zoning, cooling by steam jet, panel heating, trap maintenance. By G. D. Winans

DISTRICT heating highlights for the year 1940 were summarized in the operating statistics committee report given by Chairman W. A. Herr, Philadelphia Electric Co., at the National District Heating Association's 32nd annual meeting at the William Penn Hotel, Pittsburgh, last month. The report showed:

1) An increase in sales volume for almost all companies with the exception of several companies in the Mountain and Pacific districts.

2) An increase in revenue for practically all companies with the exception of several in the Mountain and Pacific districts and a slight decrease in average rate for almost all companies except those in the Mountain district.

3) A small increase in connected load.

4) Much colder weather for almost all the companies except those in the Pacific district.

5) An increase in cost of fuel per million Btu for most of the companies.

6) A negligible increase in number of employees, in average hourly rate and in total hourly cost of labor.

7) A slight increase in invested capital.

The first afternoon session was given over to the reports of the sales development committee (R. D. Martin, New York Steam Corp., chairman) and the commercial relations committee (George H. Tuttle, Detroit Edison Co., chairman). The sales development committee presented three inspirational speakers—George W. Ousler and T. O. McQuiston, of Duquesne Light Co. and Allegheny County Steam Heating Co., and P. C. Hodill, president of the National Association of Building Owners and Managers.

F. W. Hutchinson, University of California, gave a discussion on graphic analysis of intermittent heating for the commercial relations committee, and Dr. F. C. Houghten of the ASHVE research laboratory talked on research trends in heating.

Part of the commercial relations committee report was the discussion regarding rewriting the booklet on the principles of economical heating. This booklet was first written in 1926, and rewritten in 1933, 1937 and again in 1941. Supplementary discussion was presented by H. T. Kucera, Marsh Tritrol Co.

V. B. Ingram, New York Steam Corp., prepared a report on zoning for the commercial relations committee covering a study of 24 buildings. Analysis of operation before and after zoning the heating systems showed a reduction in steam consumption of 10 per cent for the 20 buildings of the group which obtained savings.

DISTRICT HEATING—Mr. Winans reviews here some of the highlights of the National District Heating Association's annual meeting held in Pittsburgh last month. The author is engineer of steam distribution with the Detroit Edison Co., a past-president of the NDHA and a member of the NDHA executive committee

About 20 manufacturers exhibited at the meeting, and the first evening session was given over to a visit to the exhibit, followed by a party for the delegates given by the manufacturers' representatives.

Steam Piping

The report of the distribution committee by D. L. McNulty, Allegheny County Steam Heating Co., discussed the operating problems with a high pressure steam line in Boston, and the heat loss in a 2 in. insulated steam line as affected by velocities.

The meters and accessories committee report (A. W. Wieters, Toledo Edison Co., chairman) discussed a method of testing response of differential pressure in flow meters (prepared by Thomas Ravese, Consolidated Edison Co. of New York), and condensation meter testing and maintenance cost, by C. A. Pohlig, Union Electric Co. of Missouri.

Institutional Buildings

The steam station engineering committee (A. S. Griswold, Detroit Edison Co., chairman) presented a paper on some problems in steam generation and use in institutional building groups, by A. A. Cummins and H. N. Barnard, consulting engineers. This paper showed that by proper utilization of steam for heating, additional load could be carried on existing boilers, and the authors gave the following conclusions:

It is well to investigate the boiler plant at intervals to find out what can be done under the then existing conditions to improve efficiency. These improvements may be any of the following: (1) New fuel burning equipment, (2) added water walls, (3) added air cooled walls, (4) added economizers, (5) better feed water, (6) added instruments in order to better regulate performance.

In making any improvements to old equipment, careful consideration must be given to the probable remaining life of the existing boilers in order to assure sufficient length of life for the new equipment to justify its installation.

If additional steaming capacity is required, then the decision should be made between revamping existing equipment to get additional capacity and installing a new boiler. This is a separate problem for each plant and the decision will be influenced by age of existing equipment, cost of changes as compared to the cost of new boiler, savings to be made in each case, rate of increase in plant steam load and money available for investment.

When space is available, the installation of a new boiler in an old plant, where the old boilers are becoming obsolete, presents the opportunity to begin a new complete setup which, when it is

eventually completed, will give a new installation within the same space limits but with increased efficiency and greatly increased capacity.

Careful checks show that heat control equipment can be installed in many buildings that will pay for itself in two years or less from the steam savings. These savings in most cases, we believe, will run 15 to 20 per cent and in some cases where heating is being very badly handled, will run more. This possibility is probably well recognized by most of the district heating companies but it should be a very fruitful economy to many institutional groups and industrial plants.

Where steam can be reduced from boiler pressure to heating header pressure by passing it through a turbine or engine, instead of a reducing valve, mechanical or electrical power can be secured for approximately ½ lb of coal per kwhr. There is, however, the investment and operating cost of the engine or turbine to consider, which may or may not make this practice a profitable procedure. There are a wide variety of applications to secure this benefit and a careful analysis should be made in each case.

Studies and observations point convincingly to the fact that a very large number of steam plants can cut fuel costs 15 to 30 per cent by improvements in boiler room efficiency, by controlling heating steam and in some cases by generating byproduct power.

This report was followed by a film showing stoker fuel bed operation, by C. G. Hamel and W. H. Dargan. The fuel bed was shown in fast motion, both in black and white and in colored film. Mr. Hamel said that the study of these motion pictures had already brought about changes in design in stokers.

Steam Jet Cooling

Steam jet cooling equipment, trap maintenance and panel (or radiant) heating were the subjects discussed in the research committee report. G. K. Saurwein, Harvard University, committee chairman, stated that improvement of summer load factor of district steam plants resulting from the steam requirements of cooling equipment is ample justification for any effort the committee can make to further the use of this cooling method. Simplicity of maintenance of steam jet equipment was indicated in a discussion by Richard E. Hull, New York Steam Corp. No major maintenance was required "even by a machine that had not been opened in seven years," he said, and the necessary maintenance was more in the nature of good housekeeping rather than repair or replacement of parts. If a simple program of cleaning is followed — even infrequently — no difficulties need be expected in operation, he stated.

A brief account of a special steam jet job for cooling brine to 25 F was given by L. S. Stinson, Elliott Co.

Trap Maintenance

R. Y. Sigworth, Pennsylvania State College, presented a discussion of maintenance of thermostatic traps on radiators and risers. According to his records, approximately 3300 radiator traps were in service on the Pennsylvania State campus during 1940-41, not including 11 new buildings put in operation in 1940. One hundred and eighty-five of the 3300 were replaced during the year. The record for this and the preceding year (that is, a two year period) shows a total of 280 traps repaired.

Mr. Saurwein gave a brief note on thermostatic radiator trap maintenance at Harvard, where there are about 9000 traps in service (most of the buildings are heated with hot water). Since 1930-31, when an extensive program of installing replacement parts was carried on, only 15 replacement elements have been required, and the same satisfactory service has been had with the traps installed since then. He stated that a contributing factor to this success was the installation of automatic controls and the reduction of operating pressures from 5 to 7 psi to less than 2 psi and in some cases to less than 1 psi.

Radiant Heating

T. Napier Adlam, Sarco Co., gave an extensive review of radiant or panel heating, as this method has attracted widespread attention in recent months; he discussed both principles and applications. Peter Gordon, Wolff & Munier, Inc., followed with a discussion of panel heating in the United States. He said that the most desirable location for low temperature radiating panels is at the ceiling, with floor coils for exceptional cases.

Degree-Day Base

The research committee report also included a brief discussion of the suitability of the 65 F base for the degree-day, by S. S. Sanford, Detroit Edison Co. The 65 F base is sometimes questioned, but Mr. Sanford concluded that in general, except where the normal indoor temperature is less than 70 F, it appears that the 65 F base is sufficiently accurate, and that the use of various other bases would cause confusion and make comparisons difficult.

W. F. Friend presented data on absorption refrigeration operating economies.

In the report of the rates and regulations committee (Russell Hastings, Boston Edison Co., chairman), Harry Weitzman presented a new type of rate for steam service, which is designed to encourage off-peak load.

At the last session, T. J. Finnegan, Consolidated Edison Co. of New York, gave the report of the chemistry committee, and Landis Shaw Smith, Rochester Gas & Electric Corp., gave the report for the educational committee. The educational committee this year conducted a thesis contest among the students of engineering colleges.

Officers elected for the coming year are L. S. Phillips, New York Steam Corp., president; R. M. Nee, Boston Edison Co., first vice-president; J. F. Malone, Consolidated Gas, Electric Light & Power Co., second vice-president; J. J. Schenk, Rochester Gas & Electric Corp., third vice-president; and W. A. Herr, Philadelphia Electric Co., J. M. Arthur, Jr., Kansas City Power & Light Co., and G. D. Winans, Detroit Edison Co., members of the executive committee. John F. Collins, Jr., is secretary-treasurer, with headquarters at 827 N. Euclid Ave., Pittsburgh, Pa.

Ten miles of 1¼ in. steel pipe comprise the rink cooling system

Unused Ice Plant Capacity Serves Albeth Skating Rink

WHEN an ice company builds a skating rink as an adjunct to an existing refrigeration plant, definite modifications of some of the accepted engineering practices that are commonly applied in skating rink installations may be necessary.

Albeth Ice Co., Allentown, Pa., found it desirable to depart from two of the usual practices in the lay-out of the cooling units in a skating rink constructed last fall on land adjacent to its ice plant. The main lines from the freezing tank connect with the supply and return header pipes at a point considerably off center, and reliance is placed on valves for balancing the brine flow. Likewise gravity—rather than a purge header or vacuum system—is relied on to keep air from building up in the rink coils.

A. G. Newhard, owner and operator, has operated the ice plant for 15 years, and previous to that he had for many years made a business of installing refrigerating units in stores, clubs and restaurants. His long experience in the field led him to believe that certain innovations could be made that would hold down his investment to an amount circumstances dictated without seriously affecting operating efficiency. Successful operation of the rink has indicated the soundness of his judgment.

SKATING RINK—F. S. Clark describes the outdoor skating rink built last year by the Albeth Ice Co., Allentown, Pa., to make use of extra ice plant capacity during the late fall, winter and spring months and to provide consistently good skating in a climate that normally affords it only occasionally. Approximately 10 miles of steel piping was required, made up with couplings sealed by a special commercial brine resistant sealing compound. Three-fourth inch nipples, each equipped with a valve to regulate flow of brine, were gas welded to the header pipes

The skating rink—which is of the unroofed open air type—is known as the Albeth ice rink and is designed for operation during the late fall, winter and spring months only. The ice plant's refrigerating capacity gives enough of a boost to Jack Frost so that the rink provides consistently good skating in a climate that normally affords it only occasionally.

Since the demand for ice naturally slackens during the colder months, the skating rink provides use for the ice plant refrigerating capacity that otherwise would be partly unused. Likewise, the rink provides full time employment during cold weather for delivery truck drivers and others,

who operate an ice scraper, help in the work of snow removal and perform various duties that are necessary to keep the rink in condition. During the earlier part of the past season, when the rink was still uncompleted, the regular crew of the ice plant was even called on to work overtime, as the company itself did a large part of the construction work involved, including installation of piping.

The ice plant is equipped with an ammonia compressor of the heavy duty type, powered by a 100 hp diesel engine. Since the rink is of sizable dimensions—85 ft by 185 ft —it was not anticipated that this unit would be sufficient to hold the ice during a long skating season. Hence, provision has been made in the type and arrangement of fittings, as well as in the layout of the cooling units, for eventually installing an additional compressor. The short skating season afforded by the existing refrigeration plant was so successful that it is now planned to install a larger unit, which will make it possible to provide at least six months of open air skating.

There were a number of considerations that entered into the decision to locate the header connections as near as feasible to the ice plant rather than midway between the ends of the rink. In the first place, the point at which the connections were made, about a third of the rink's length from the end closest to the plant, is the shortest route from the freezing tank. It was felt that, while there might be some advantage in a more central location of the header connections, any gain from this would be more than offset by the numerous disadvantages of a longer connecting line to the freezing tank. Likewise, plans for installing the new compressor entered into the reckoning. A connection with the existing plant a third of the way from the end left the preferable middle location free and clear for connections with the proposed new plant. Due to his long experience in the refrigerating field, Mr. Newhard had few doubts of the practicability of the off-center location of the main supply and return connections. Adjuster valves, he believed, if properly set should balance the brine flow just as effectively as in a more perfectly balanced arrangement of the system.

Header supply and return pipes consist of 8, 6 and 4 in. steel pipe with each size extending respectively one-third of the rink's length, the step downs in diameter proceeding from the direction of refrigerant supply. Outflow and return lines of the cooling coils are respectively connected to the supply and return headers by ¾ in. steel pipes, each equipped with an adjuster valve for balancing brine flow. The condition of the ice in the rink shows which valves need adjustment. The operating experience has been that, once the valves are properly set and the brine flow balanced, no further adjustments are necessary during the skating season.

Approximately 10 miles of 1¼ in. steel pipe was used. This pipe is an open hearth grade of steel pipe made especially for refrigeration and the carrying of ammonia. In order to assure freedom of any defects in the pipe itself, it is tested with air pressure under water. To safeguard against leakages through faulty connections, special care was taken in installing the coils and other piping. Rather than sealing all connections with a litharge compound, however, a special commercial brine resistant sealing compound was used, which permits ready uncoupling of any connections that may be faulty Only 3 gal. of the compound was required.

After the installation was completed the pipes were covered with water and the system was tested against leakage under 35 psi air pressure. Only four leaks developed when the pressure was applied to the 10 mile piping system, and these in each case were due to excessive make-ups at the couplings. Since the sealing compound employed permitted ready uncoupling, these few leaks were easily remedied.

Each nipple has a valve to regulate brine flow

Photos courtesy Bethlehem Steel Co.

The self-venting action of the system is achieved just as it might be in a hot water heating installation. Flow lines for any air that may get into the system are maintained at successively higher levels. In the supply header, which is 8 in. below the rink floor cooling system, the brine reaches its lowest level. Thus an elevation is afforded in the flow to the coils. The 8 in. return header is centered at the same level as the coils, but provides a higher ceiling for air in the system to rise to, due to its larger diameter. Air from the return header can escape from the system from the open freezing tank. In order to bring the supply line to below the rink floor cooling system level, an offset bend of 3 ft was made in the supply line, which was accomplished by heating and bending the pipe itself. No purge header or vacuum system is necessary to remove air from the system because none can collect in it.

The ¾ in. nipples that connect the supply and return headers with the rink cooling coils are connected to the header pipes by gas welds. Distortion in the header pipes that resulted from uneven heating of the pipe in the welding process was corrected by spot heating the pipe with large torches and re-straightening it wherever necessary.

Operation of the rink last winter amply demonstrated that the system can be operated at a cost which is well in line with the requirements of such a business. Computations made on the basis of last winter's experience indicate that the additional equipment needed to prolong the skating season can also be installed at a reasonable cost and a much lower capital investment than would be necessary to roof the rink and rely entirely on present cooling capacity.

The rink is situated on the main highway connecting Allentown with nearby Bethlehem—hence the name Albeth. A. S. Newhard, son of the owner and operator, acts as assistant manager and helped his father in working out the design and plans.

How Air Conditioning Aids Defense

Speeds Production of Essential Airplane Instruments at the Kollsman Instrument Div., Says Edward T. Murphy

AIDS DEFENSE—Air conditioning is essential to efficient production for national defense in numerous ways, as has been pointed out practically every month in HPAC by means of specific examples. At the Kollsman Instrument Div., a new installation providing constant temperature and humidity conditions comprises a 15 ton system and a silica gel dehydrator. Mr. Murphy, vice-president in charge of marketing of Carrier Corp., explains the need for air conditioning in turning out precision instruments

SPEEDING up the production of airplane instruments needed in the national defense program will be an important result of the installation of air conditioning equipment at Kollsman Div. of Square D Co., Elmhurst, Long Island. In delicate operations required to turn out precision aircraft instruments, constant temperature and humidity conditions are of considerable importance. In order to maintain these conditions in parts of the Kollsman plant not previously air conditioned, a new 15 ton air conditioning system and a silica gel dehydrator were installed.

One of the important jobs of this installation will be the maintenance of a low humidity of not more than 35 per cent in the assembly room. The importance of maintaining this low humidity arises from the fact that the slightest perspiration on the hands of the employees handling the delicate instruments tends to cause tarnishing or rusting of the metal parts. Some of the parts are so precisely machined that even the slightest blotch of oxidation might affect the accuracy of the instrument. The removal of dust from the air will also be an important advantage.

Another reason for maintaining this low humidity is to reduce the condensation of moisture on the instruments, when they are removed from the test chambers where they have been held at —40 F. The silica gel unit will remove more than 50 gal of moisture from the air in the assembly department during an average 8 hr day.

The use of the air conditioning in the inspection and testing departments will also have the advantage of keeping the temperature at a constant figure. This is extremely important, as the tolerances in many cases are so close that a slight variation in temperature has a tendency to make the instrument vary from the standard and therefore make accurate testing more difficult.

Aside from the benefits derived from the air conditioning in its effect on the manufacturing processes, it is expected that the resulting increased efficiency on the part of the employees due to improved comfort conditions will also result in greater production.

Plant Protection—A plant protection inspection service has been organized by the war department to assure the uninterrupted production of the materials necessary for national defense, according to an announcement last month.

The service has been organized to protect industrial plants that are either plant contractors or subcontractors on army orders against damage that might cause stoppage, interruption or retardation of production schedules. Particular emphasis is put on the control of the fire hazard.

The inspection force consists of officers and civilians. The war department states that great emphasis will be placed on selecting only men of the highest quality and that emphasis will be placed on obtaining men of inspection experience in insurance fields or industrial engineering.

The features of plant protection cover such matters as adequate protection against the ever-present and highly dangerous fire hazard, identification of employees upon their entrance to the plant premises, and control of visitors.

IT'S A JOB—Advancement in the art of air conditioning and the ever-growing appreciation of what it means to a building leads to more and more existing structures installing equipment. But older buildings, erected before air conditioning became more or less "standard practice," often don't have the space for the air treating equipment and the ducts. Ingenuity on the part of the designers and installers is therefore required. . . . In the three story bank building that Mr. Lowell (air conditioning engineer) describes here, a false panel was built on each side of the semicircular revolving door casing, and the space is used to carry the air from the equipment in the basement to the top of the door, where it is diffused through a directional flow register. An excavation under the sidewalk provided space for the apparatus. . . . Well water is the cooling medium

IT'S A JOB

sometimes to find the space in existing buildings for air conditioning equipment. W. E. Lowell tells how the problem was solved when a three story building modernized

NECESSITY is the mother of invention—this is particularly true in regard to the problems that frequently confront the air conditioning engineer.

In new structures, where air conditioning is contemplated as an immediate or future installation, ample space for the equipment is usually—but not always—allotted by the architect or structural engineer. But since the art has advanced to a point where scarcely an industry or building is not benefited in some measure by air conditioning, finding a suitable location for the system in existing structures is a problem that comes up more and more.

A case in mind is the First National bank building in Elkhart, Ind. The first floor, mezzanine and a portion of the third floor is occupied by the bank, and the balance of the third floor is rented. When this structure was erected the possibility of air conditioning it was as remote as passenger service to the moon.

However, a year ago it was planned to install a system, an appropriation was made to cover the cost, and design of a system that would produce the desired results yet not detract one iota from the architectural beauty and symmetry of the bank interior was specified. It was decided that well water would be the cooling medium, as it is readily obtainable in sufficient quantities and at proper temperatures at a depth of 135 ft.

That matter settled, the next question was location of the air conditioner. The logical place was the basement, but no possible space could be found. Furthermore, the ceiling was low and covered with a myriad of pipes; also, the space was divided off by several heavy masonry walls and contained a coal bunker, two vaults, boiler room and a toilet room. There was no straight passage through from one end to the other. There just wasn't any space in the basement for the air conditioner—and if there had been, it would have been virtually impossible to thread a duct system through except at an unwarranted expense. In addition, building inspectors and architects are very reluctant to have bearing walls hacked to pieces.

A blank floor plan was used to lay out an ideal air distributing system and then a survey was made to determine if it could be fitted in; but that idea was abandoned because it

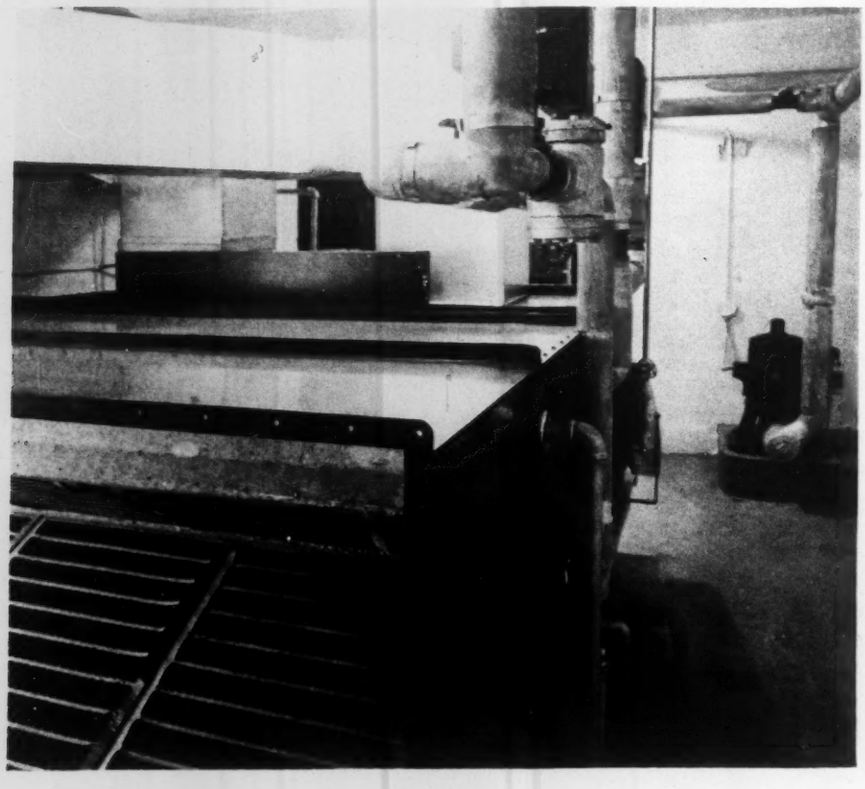
meant cutting a 3 ft by 2 ft opening at each end of a fireproof vault in which valuable records were stored and would have brought the ducts up against the breeching of the boiler. Suggestion upon suggestion was offered as a possible location, but each in turn was discarded because of lack of space, disfigurement of the building or excessive cost.

As a final solution to the problem it was decided to use the space immediately adjacent to the revolving door casing. This door is about midway on the long side of the building (the building is 161 ft by 21 ft) and the top makes a good distributing point for the air in all directions. A false panel was made on each side of the semicircular door casing and the space in between (about 8 in. by 60 in.) is used to carry the air from the basement to the top of the door where it is diffused through a vertical directional flow bronze register. The return air is brought back through bronze grilles at the floor level in the corners of the door casings and the wall. These tie into the outside air ducts, the registers of which are on each side of the entrance to the revolving door and about 8 ft above the sidewalk. The return air registers were provided with dampers so that a complete control of outside and return air can be had.

In the basement, directly under this door, there had at one time been a window pit, the opening of which had been filled in with brick. An excavation was made under the sidewalk to house the air conditioner, and this opening was used to accommodate the ductwork to and from the revolving door. With this arrangement we were able to make the whole installation with about 15 ft of supply duct and 4 ft of return duct, a minimum amount of cutting and patching in the bank proper, and with no changes to the building structure. No space was taken from the building by this procedure except 8 in. on each side of the revolving door. However, while making the excavation under the sidewalk, we went beyond the normal requirements for the air conditioner and excavated an additional space to provide a "dead" storage vault.

The second floor, mezzanine and top floor are served by a unit in a storage room at one end of the top floor hall. The waste water from this unit is discharged onto the roof through spray nozzles to reduce the solar heat load.

Seen here is the well pump and the filter section of the air conditioner, which is installed in excavated space under the sidewalk



CONDITIONING BOMBPROOF UNDERGROUND SPACES

The conditioning of bombproof underground spaces for shelters and factories is discussed by Edward Ledoux in the June, 1941, Chemical and Metallurgical Engineering. He was engaged in design of such spaces before the occupation of France.

Owing to heat capacity of masonry and earth, underground bombproof spaces remain at almost constant temperature (50 to 60 F in temperate zone) throughout the year and the air is nearly always saturated and frequently supersaturated. Any one of three methods of removing moisture from the air —refrigeration, absorption or adsorption—may be used, the particular type depending on circumstances.

Evaporation of water from the surfaces of porous masonry is generally the most important source of moisture. The coefficient of vapor transfer is least in the stationary air at the wall surface, and this emphasizes the need for disturbing this air as little as possible through minimizing drafts and use of low register velocities.

Underground storage rooms generally are of large internal volume with occupancy so small as to be negligible. Conditioning in this case is for the purpose of preventing condensation, and the temperature, within limits, is immaterial. Adsorption with silica gel or activated alumina gives the desired solution to the problem because of the extremely low dew point obtainable, because low temperatures favor adsorption, and because cooling water is not necessary. Shelters—except for protection against gas attacks—present quite a different problem from that of storage spaces owing to the presence of persons.

Two distinct methods are used for ventilating gasproof shelters. In the closed circuit method, breathing requirements are met either by regeneration of the internal air, with carbon dioxide absorption and oxygen supply, or by expansion of compressed air stored in high pressure cylinders which are recharged by a compression plant between attacks. In the open circuit system, air is introduced from outside, after stripping it by proper methods of all toxic substances.

EXPAND THE USE OF FUEL OIL

Says Oil Heating Authority

KALMAN STEINER DISCUSSES THE FUEL OIL SITUATION, BELIEVES OIL USE SHOULD BE EXPANDED RATHER THAN CURTAILED . . . . . OPERATING ENGINEERS SHOULD COOPERATE IN CONSERVING NATION'S FUEL RESOURCES

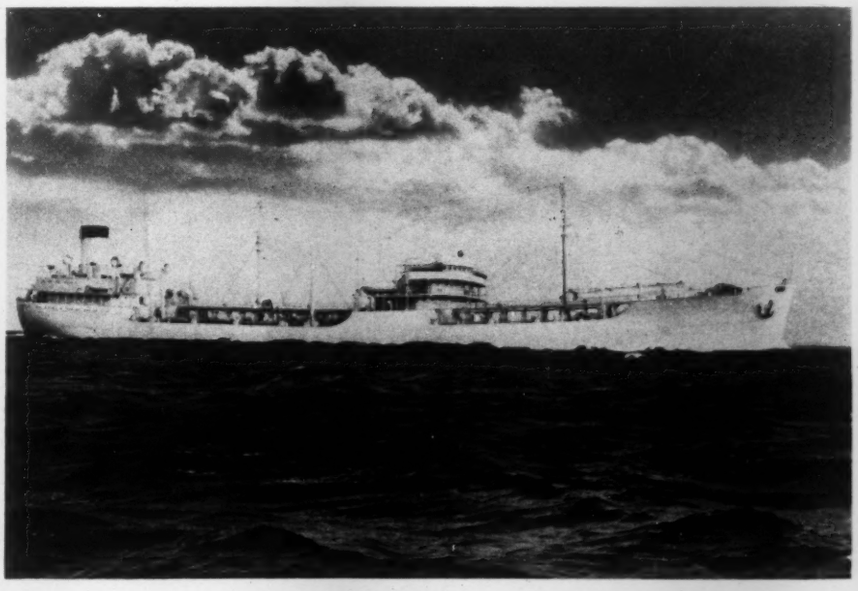
NEWSPAPER headlines of the past two or three months convey the impression that our defense program and the unlimited emergency proclaimed by President Roosevelt will create a severe shortage of fuel oil—or at least that an acute and sudden transportation deficiency will greatly reduce the supply available in 13 states of the Atlantic Seaboard. In fact, early in May an official of the Office of Price Administration and Civilian Supply issued from Washington a request that oil burner manufacturers desist from selling new installations east of the Appalachians. Subsequently, that suggestion was modified considerably, but at least it is obvious that some unusual situation must suddenly have been created to have evoked even the suggestion of such a plan. Heating men naturally are concerned with fuel oil developments, so readers of HPAC no doubt will find profitable (or at least, enlightening) a brief review of the facts, particularly with regard to the fuel industry as a whole. Fuel oil, like all fuels, bears an important relation to industrial activity during a national emergency. I hope to show in this discussion that the use of fuel oil should be expanded rather than curtailed at this time.

65 Per Cent of World Supply

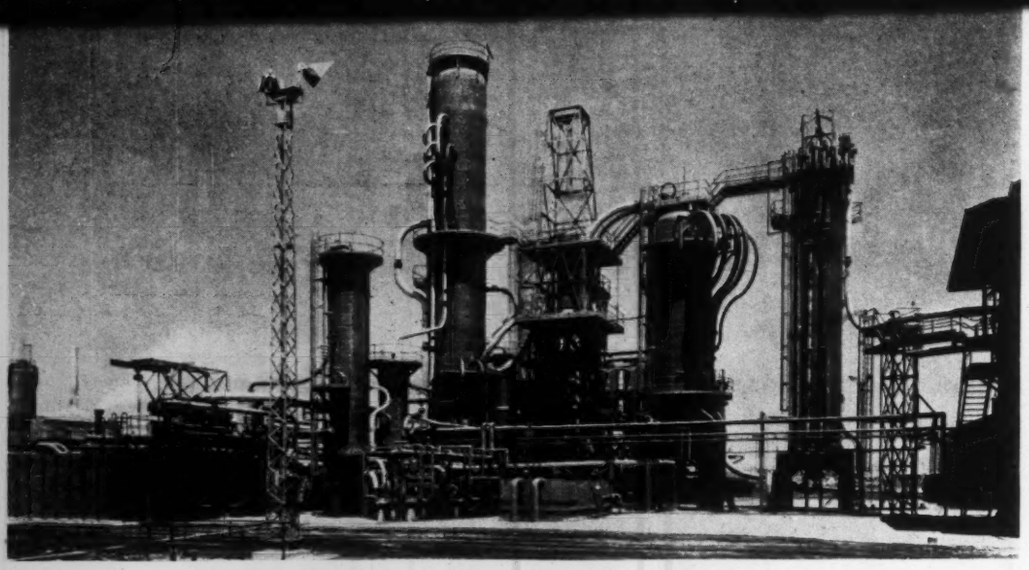
The United States produces roughly 65 per cent of the world's petroleum supply. Our Eastern Seaboard consumes nearly half of our total output, and so requires about one-third of the world's annual production. For those who enjoy statistics, this East Coast consumption can be stated as an average of 1,300,000 bbl (1 bbl = 42 gal) a day or some 474,000,000 bbl a year. This includes all petroleum products —gasoline, kerosene, naphtha and solvents, lubricants and greases, light and heavy fuel oils. Since most of this material comes from American wells in the Southwest, and since the least expensive way to transport it to the East has been via the sea route with tankers, a fleet of 325 tankers had been built up to handle this traffic. Over 90 per cent of all petroleum products used along the Atlantic has been brought there by tankers.

Transportation Shortage

The sudden emergency that stirred up the wave of press reports on fuel oil shortages was induced by our giving Britain 50 tankers, thereby reducing our available marine transportation facilities by, roughly, 15 per cent. Of course, there is no assurance that we will not give away still more—but neither is there certainty that we will. Besides the original 325 tank-



FUEL OIL—Transfer to Britain of tankers supplying our Atlantic Seaboard with petroleum products has caused widespread speculation on the possibilities of fuel oil shortages—not only in the East (the area affected) but in other sections of the U. S. as well. Mr. Steiner (of Ace Engineering Co.), author of a number of articles which have appeared in HPAC and of a well-known text on oil burners, analyzes the situation here and the steps being taken to correct it. He concludes that increased use of fuel oil should receive favorable consideration, especially in the Middle West, and explains his reasons. . . . There is no doubt, however, that all should assume the responsibility of helping to conserve the nation's fuel resources. Operating engineers can do this by achieving more efficient combustion of fuel and more effective utilization



Essentially, fuel oil is a byproduct of gasoline manufacture

ers plying the Atlantic and Gulf Coasts, we also have another hundred or so working the Pacific Coast. England started the war with a tanker fleet of 435 of her own registry and 63 under dominion flags. Some of the British ships, of course, have been sunk, but many of the 379 Norwegian and Dutch tankers have now become part of Britain's fleet. Numerous new tankers are under construction, and announcement has been made that certain Pacific Coast shipyards will immediately devote their efforts exclusively to constructing tankers—larger and faster ones. The figures on tankers therefore do not look too despairing.

Abundant Stocks

There are abundant stocks of crude oil and refined products to serve our needs, and the proved resources still underground are greater than at any previous time. No question of shortage has been raised. The problem is purely one of transportation, and that problem affects only the East Coast. But it relates equally to all petroleum products—not merely to fuel oil. Already Mr. Ickes, oil industry coördinator, has asked the people of that area to curtail their gasoline consumption by omitting Sunday driving, by reducing driving speeds in the interest of improving mileage per gallon, and in general eliminating waste. That a 15 per cent reduction in gasoline consumption can be accomplished through coöperative effort of this kind can almost be taken for granted.

Reducing Consumption

On its part, the Oil Burner Institute has given Mr. Ickes its pledge to undertake a sweeping campaign to reduce fuel oil consumption. Through its membership of oil burner manufacturers and their dealers a six point program has been instituted, encompassing (1) a war on waste, (2) an effort to bring possibly 2,000,000 oil burner installations up to present-day operating efficiencies, (3) installation of all new burners in strict conformance with the Bureau of Standards commercial standard CS 75-39, (4) improvement of building insulation, weatherstripping, etc., to save fuel, (5) coöperation with other branches of the petroleum industry, and (6) conservation of use. With such a coöperative program carried into effect, the oil burner industry should easily effect its share in the required minimum reduction of 15 per cent in consumption.

The remainder of the United States is independent of the tanker situation. We have a network of 117,000 miles of pipe line conveying crude and 10,000 miles of gasoline pipe line. In addition, there are 147,919 petroleum tank cars in service, which during 1940 transported 60,000,000 tons of petroleum products. Finally, thousands of transport trucks are used for delivery of finished products from refineries.

Despite this abundance of transportation means, and the vast available supply of crude and refined products, the price of fuel oil at inland points has been advancing and there has been some reluctance on the part of refiners to take on new commitments for fuel oil delivery during the coming season. Two facts may be ascribed in explanation: prices of practically all commodities have been advancing and there is no reason why fuel oil should be an exception, and the operation of steel mills at practically 100 per cent capacity, and greatly increased industrial activity generally, absorbs tremendous quantities of fuel oil.

Nation's Fuel Requirements

There is still good reason to believe that oil should continue as a fuel of major importance and that its use should be extended, in my opinion. First, consider the alternatives. Assume for a moment that other fuels were to be generally substituted where oil is now burned. The country produced, roughly, half a billion barrels of fuel oil during 1939.* Of this, 17.4 per cent was burned on steamships, 14.7 per cent in metallurgical and other industries (where presumably the process cannot readily be made coal burning), and 2.7 per cent by the navy, army and coast guard, or a total of 34.8 per cent—leaving 65.2 per cent or 325,000,000 bbl that could conceivably be replaced by some other fuel.

Our total production of natural gas during 1938,* converted on an equivalent Btu basis without regard to differences of efficiency (if any), was equal to 386,000,000 bbl of fuel oil, so that to replace oil fuel with natural gas would require doubling present gas production rates, to say nothing of doubling the mileage or at least the capacity of the present natural gas distribution system of 190,000 miles of pipe line.

Converting 325,000,000 bbl of fuel oil to coal, also on straight Btu comparison, requires 81,000,000 tons, or about one-fourth of our total 1938 coal production. But since oil is utilized at greater efficiency than coal, especially in millions of domestic heating plants, it is more likely that we would have to increase our coal production by 40 per cent to replace present oil burning plants with coal burners. Normally, our railroad traffic is composed to the extent of 37 per cent of coal and coke. Complete substitution of coal for oil would therefore increase rail freight by 15 per cent and the railroads now face an enormous problem of moving defense materials. The total coal, bituminous and anthracite, moved into New England during 1938 was 17,500,000 tons. The total fuel oil moved into New England during 1939 was 46,481,000 bbl, which in straight Btu is the

*All statistics are the most recent easily available.

equivalent of 11,000,000 tons of coal, but in actual utilization the equivalent of probably 20,000,000 tons of coal. To replace all fuel oil, therefore, the railroads would have to more than double their coal traffic to New England. And finally, the existing distributing channels of bulk plants and tank trucks would have to be promptly replaced by coal yards and coal trucks.

Pipe Lines and Barges

These figures show rather convincingly, I think, that so far as East Coast fuel is concerned the *status quo* will prevail: First, because there will be sufficient fuel oil there and, second, because no substitute fuel is feasible. Immediate steps should be taken, of course, to alleviate the condition of complete dependence upon marine transportation. Two alternatives present themselves: Pipe lines extending from the Southwest to the Atlantic Coast, and barge movement through inland waterways eastward as far as possible, possibly to Pittsburgh, supplemented by pipe line for the remaining distance. Congress has already passed legislation toward expediting pipe line construction for the entire distance, and rapid strides may be expected in the laying of new pipe lines almost immediately. But because pipe line transportation is somewhat more expensive than water transportation, thought should also be given to inland waterways and barges. The Mississippi and Ohio river routes may prove to be an ideal solution to eastward movement of petroleum.

Increase Use of Oil

I stated above that I believe increased use of fuel oil should receive favorable consideration at this time.

To replace all fuel oil, railroads would have to more than double their coal traffic to New England

This applies especially to the Middle West. Despite the greater industrial demand for fuel oil created by acceleration of the steel and other industries of the Middle West, it must be remembered that fuel oil is essentially a byproduct of gasoline manufacture and that modern defense preparation must be thought of in terms of gasoline propelled machines: airplanes . . . tanks . . . trucks. As our gasoline production is stepped up to meet this defense requirement, there will become available increasing quantities of fuel oil. This fuel oil cannot readily be stored in quantities approaching actual production, but must be consumed at about the same rate as produced. It can be expected therefore that within a short time refineries will catch up with—and then exceed—the present fuel oil demand, thereby tending to depress the price.

The extended use of fuel oil is important from another angle. Fuel oil involves but little manpower in either its production or consumption, and so helps to conserve that greatest of our national resources—manpower.

How to Cooperate

But that all of us should feel a responsibility in helping to conserve our fuel resources, there can be no doubt; both Mr. Ickes and the OBI have indicated the urgency of everyone's coöperating. The operating engineer can do much. His efforts should be directed toward both more efficient combustion of fuel oil and more effective use of heating plants. Numerous articles in HPAC have stressed these points, among which the writer has contributed a few. Future articles will recapitulate the essential points of what the operating engineer can do to assist in fuel conservation.

[Editor's Note: In the meantime, see the following articles by Mr. Steiner, which have appeared in HPAC: How to Operate Oil Fired Heating Plants Efficiently (with Paul R. Unger), May, 1936; Combustion Control for Oil Fired Boiler Plants (with Paul R. Unger), October, 1936; Heating System Control Methods for Reducing Fuel Consumption, February, 1937; Heating System Control Methods and Reducing Fuel Consumption, March, 1937; Regular Supervision of Apartment Hotel Heating Plants Cuts Costs, July, 1939.]

CHECK PIPING FOR SAVINGS

A few years ago a survey was made in a plant to determine the extent of the indirect losses and to see what might be accomplished in reducing condensate loss. Fortunately the plant was not operating full time and it was possible to do a thorough job. Traps were inspected and repaired, drain lines were checked and valves removed and overhauled. Every major piece of equipment using steam was tested. Leaks were discovered in several large heating coils. Many traps were not functioning.

As an example of lack of attention, the drains from two large blast coils had been disconnected during a repair job some years previous and permitted to discharge temporarily into the sewer. This temporary arrangement had been forgotten and during the intervening period of several years all the condensate had been lost. At the return pump a continuous stream of cold raw water, more than was required for make up, had been injected into the condensate line to kill the vapor.

Make Up Reduced

When the job was completed the percentage of make up water for boiler feed had been reduced from approximately 60 per cent to less than 40 per cent. In this plant a large storage tank for treated water was later installed to be drawn on for starting up periods.—R. S. Hawley, chairman of the department of mechanical engineering, University of Michigan, and member of HPAC's board of consulting and contributing editors, at the Midwest Power Conference.

Steel Socket Welding Fittings—A draft of a proposed American standard for steel socket welding fittings has recently been completed. This proposal covers overall dimensions, tolerances and marking for wrought and cast carbon and alloy steel welding fittings. It is now being distributed to industry for criticism and comment, and copies are available from C. B. LePage, assistant secretary of the American Society of Mechanical Engineers, 29 W. 39th St., New York, N. Y.

THE WATER-AIR CHART

Its Use As a Psych Chart.....By William Goodman*

WATER-AIR CHART — The water-air chart provides a graphical method by which a great many types of problems dealing with the exchange of heat between air and water can be greatly clarified and conveniently solved for numerical answers. It can be used not only for problems involving the exchange of heat between air and water in spray chambers and coils, but also for problems dealing with the temperature of the spray water in evaporative condensers and spray coolers, and for problems dealing with the surface temperatures of coils. . . . It provides a particularly clear picture of the action taking place in air washers, cooling towers and coils. With its aid many difficult problems can readily be visualized and solved. This is particularly true of counterflow problems, otherwise difficult to visualize. . . . The concluding article of the series on the water-air chart published this month covers its use for dry surface cooling; the beginning material on the use of the water-air chart as a psychrometric chart is needed in order to understand *how* to use the chart for dry surface cooling. Thus, the chart is universal in its application. It can be used for dry surface surface cooling, wet surface cooling or cooling in which dry and wet surface occur

THE water-air chart which was published on p. 222 of the April HPAC may be used as a psychrometric chart. In this case, the scale of temperature on the chart may be used for either dry bulb, wet bulb or dew point temperatures as is the case for the psychrometric chart published in HPAC, June, 1939. Thus, in the small-scale sketch of the water-air chart shown in Fig. 20, point *1* is located at the intersection of the dry bulb temperature and the enthalpy of the air.

To use the water-air chart as a psychrometric chart, lines of constant specific humidity are required. These lines of constant specific humidity can easily be drawn as needed because they are straight lines on the water-air chart. However, these straight lines of constant specific humidity are not parallel to each other, although they are very nearly so. The slope of a line of constant specific humidity is given by the following equation (which is derived in the section at the end of this article headed Slope of Dew Point Lines):

$$N = c \quad \ldots\ldots\ldots\ldots\ldots\ldots [17]$$

$$c = 0.24 + 0.45w \quad \ldots\ldots\ldots\ldots [18]$$

Thus, according to equation 17, the lines of constant specific humidity are straight lines and their slope is equal to the humid specific

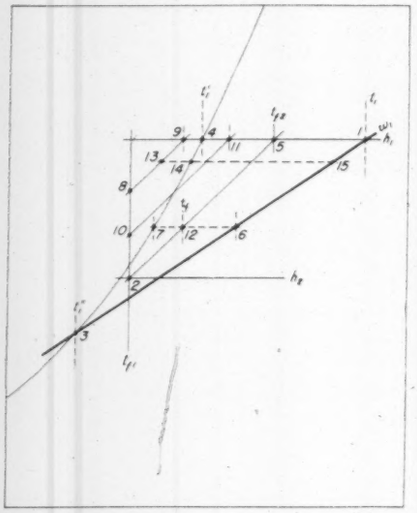


Fig. 20—Counterflow cooling with dry surface

heat c. The value of c can be obtained from the psychrometric tables (HPAC, January, 1938, pp. 4-6) or computed by equation 18. From equation 18 it is also apparent that the slope of each line of constant specific humidity depends upon the value of w for which the line is drawn. Knowing the value of c for any given line of constant specific humidity, the angle θ can be obtained from Table 2 (HPAC, April, 1941, p. 223) after first locating the value of c in the column headed N.

A line of constant specific humidity can also be located on the water-air chart by locating the point corresponding to the initial enthalpy and dry bulb temperature. Then the dew point temperature as read from the psychrometric chart published in HPAC for June, 1939, can be located on the saturation curve of the water-air chart. The straight line between these two points is the line of constant specific humidity corresponding to the initial specific humidity of the air. Thus, in Fig. 20, point *1* is located at the intersection of the enthalpy and the dry bulb temperature of the air. Point *3* on the saturation curve is the dew point temperature corresponding to point *1*. Line *1-3* between the two points is the line of constant specific humidity corresponding to the specific humidity of the air in state *1*.

In Fig. 20, point *1* represents the initial enthalpy and dry bulb temperature of the air, and line *1-3* the initial specific humidity of the air. If air is cooled at constant specific humidity, the final state of the air must be represented by a point on line *1-3*. Thus, if the air is cooled to point *6*, the final dry bulb temperature of the air at point *6* is read on the vertical line through *6*. The final wet bulb temperature of the air is read at point *7* where the horizontal line through point *6* intersects the saturation curve. The dew point temperature, of course, remains constant at point *3*. The wet bulb read at point *7* is approximate, of course, but is very nearly equal to the correct wet bulb temperature. The exact wet bulb temperature at point *6* can be found by referring to the psychrometric charts published in HPAC for June, 1939.

Symbols

c = humid specific heat of air (c = 0.243 approximately).
h = enthalpy of saturated air at wet bulb temperature t'.
N = see Table 1, p. 221, April, 1941, HPAC.
t = dry bulb temperature of air, F.
t' = wet bulb temperature of air, F.
t'' = dew point temperature of air, F.
t_t = temperature of liquid, F.
w = specific humidity of air, lb of moisture per lb of dry air.
θ = angle for N-line.

Numerical subscripts refer to condition of air or liquid at various points in heat transfer process. For example, subscripts 1 and 2 refer respectively to initial and final conditions of air or liquid.

Dry Surface Cooling

In discussing the application of the water-air chart to problems in which the water is separated from

*Consulting engineer, The Trane Co. Member of HPAC's board of consulting and contributing editors.

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the air by a metallic surface, only the case in which the specific humidity of the air changes has been considered heretofore. The specific humidity of the air can change only if the air is in contact with a film of moisture on the metal surface. On the other hand, if the surface is dry, the specific humidity of the air will remain constant while the air is being heated or cooled.

The water-air chart is also applicable to dry surface problems in which the specific humidity of the air remains constant while the temperature of the air changes. Problems involving dry surface are solved in exactly the same manner as problems in which the specific humidity of the air changes. Thus, the *N*-lines for parallel flow heating and cooling at constant specific humidity are similar in appearance to the *N*-lines of Figs. 4 and 6 respectively (see HPAC, April, 1941, p. 224 and May, 1941, p. 291). For problems in counterflow heating and cooling at constant specific humidity, the *N*-lines are similar in appearance to the *N*-lines of Figs. 8 and 9 respectively (see HPAC, May, 1941, pp. 292 and 293).

For dry surface cooling or heating, the *N*-lines are always drawn between the points located at the intersection of the enthalpy of the air and the temperature of the water exactly as is the case when the specific humidity of the air changes. However, the limiting *N*-lines for dry surface cooling or heating are altogether different from the limiting *N*-lines for wet surface cooling or heating. When the surface is dry, the transfer of heat between the water and air is governed altogether by the dry bulb temperature of the air—not by the wet bulb temperature, as is the case when the air is in contact with a wet surface. Consequently, the limiting case is reached when the *dry bulb* temperature of the air is equal to the temperature of the water. As is shown below, the limiting *N*-line for dry surface cooling or heating coincides with the line representing the initial specific humidity of the air.

As air is cooled, the final dry bulb temperature of the air is read along the line of constant specific humidity *1-3* of Fig. 20. Now suppose that line *1-3* is regarded as an *N*-line. At any point on this *N*-line the temperature of the water corresponding to the enthalpy of the air may be read, as is possible for all other *N*-lines. Thus, when the air is cooled to point *6*, the temperature of the water can be read on the vertical line through point *6*. However, the dry bulb temperature of the air is also read on the vertical line through point *6*. Evidently, at any point along the line corresponding to the constant initial specific humidity of the air, the temperature of the water is equal to the dry bulb temperature of the air. Consequently it is apparent that the line of constant specific humidity represents the limiting *N*-line along which the exchange of heat between the water and air may theoretically take place. For the exchange of heat between the air and water through dry surface, the limiting *N*-line is not determined by the saturation curve as is the case when a change in the specific humidity of the air takes place. For dry surface cooling or heating, the limiting *N*-line coincides with the line representing the initial specific humidity of the air.

Although the limiting *N*-line is represented by *1-3* in Fig. 20, the actual cooling of the air will take place along an *N*-line such as *8-9*, *10-11*, or *2-5*. *N*-lines located similarly to line *8-9* have been discussed in detail in earlier sections of this series. However, an *N*-line such as *2-5* needs further explanation. If the corresponding enthalpy of the air and the temperature of the water are represented along a line such as *2-5*, it simply means that the temperature of the water is always between the dry and wet bulb temperatures of the air. The dry bulb temperature of the air is read at point *6* and the wet bulb temperature at point *7*. The temperature of the water is represented by the vertical line through point *12*. Consequently, it is apparent that the water temperature at any point along line *2-5* must be between the dry and wet bulb temperatures of the air. On the other hand, at any point such as *13* on an *N*-line like *8-9*, the temperature of the water is always between the wet bulb and dew point temperatures of the air. On an *N*-line like *10-11*, the water temperature will be between the dew point and wet bulb temperatures of the air for part of the process, and between the wet bulb and dry bulb temperatures of the air for the remaining part of the process.

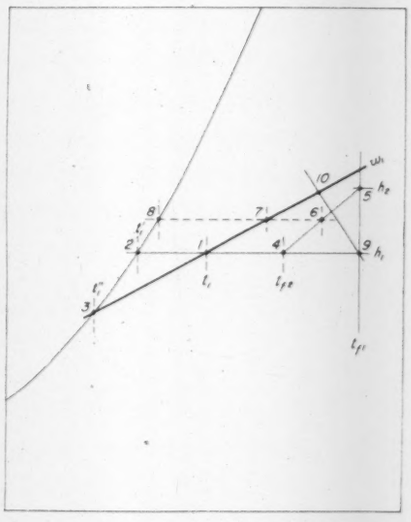


Fig. 21—Counterflow heating and parallel flow heating with dry surface

In all cases, when cooling of the air takes place at constant specific humidity, the water temperature must always be lower than the *dry bulb* temperature of the air. Furthermore, the dry bulb and wet bulb temperatures of the air and the temperature of the water always lie on the same horizontal line. Thus, for point *12* of Fig. 20 the dry bulb temperature is read at point *6*, the water temperature at point *12*, and the wet bulb temperature at point *7*. For point *13*, the dry bulb temperature of the air is read at point *15*, the wet bulb temperature at point *14*, and the water temperature at point *13*.

In Fig. 20, constant-specific-humidity line *1-3* should not be confused with an *N*-line such as *2-5*. Line *1-3* represents the actual change in the state of the air. In other words, the dry and wet bulb temperatures of the air at any time can be read along line *1-3*. On the other hand, *N*-line *2-5* represents only the relationship between the enthalpy of the air and the temperature of the water. It does not at any time represent the dry bulb temperature of the air.

In dry surface counterflow heating, the points through which the *N*-line must pass are also determined wholly by the enthalpy of the air and the temperatures of the water in the manner illustrated in Fig. 8 (p. 292, May, 1941, HPAC). However, the limiting *N*-line for this case also coincides with the line representing the initial specific humidity of the air. Thus, in Fig. 21 point *1*

is located at the intersection of the initial dry bulb temperature of the air and its initial enthalpy. Point *3* is located where the initial dew point temperature of the air intersects the saturation curve. Line *1-3* is a line representing the initial specific humidity of the air. It also represents the limiting *N*-line. Actual heating of the air would take place along an *N*-line such as *4-5*. If the air were heated to point *6*, the temperature of the water would be read at the vertical line through point *6*. The

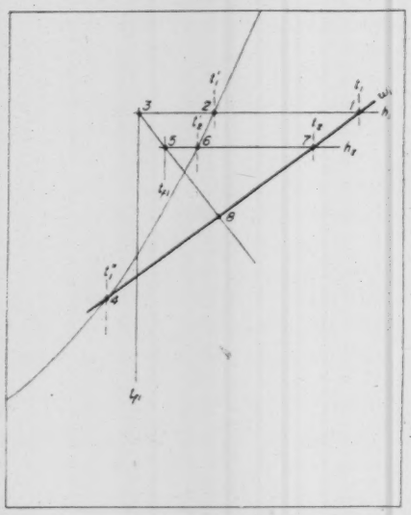


Fig. 22—Parallel flow cooling with dry surface

dry bulb temperature of the air would be read at point *7* and the wet bulb temperature of the air at point *8*. When heating air, the temperature of the water is, of course, always higher than the dry bulb temperature of the air.

Parallel flow heating of air by dry surface would be represented by a line such as *9-10*. The theoretical limit of such heating is represented by point *10*, where the final dry bulb temperature of the air is equal to the final temperature of the water.

Parallel flow cooling of air by dry surface is illustrated in Fig. 22. Point *3* is located at the intersection of the horizontal line representing the initial enthalpy of the air with the vertical line representing the initial temperature of the water. Line *1-4* is the line of initial specific humidity of the air. The relationship between the enthalpy of the air and the temperature of the water at any point in a parallel flow cooling process is represented by a point on an *N*-line such as *3-5*. If the air is cooled to point *5*, the temperature of the water is read at point *5*; the wet bulb temperature of air, at point *6*; and the dry bulb temperature, at point *7*. The dew point temperature of the air will, of course, be constant and represented by point *4*. The theoretical limit of such cooling is represented by point *8*, where the final dry bulb temperature of the air is equal to the final temperature of the water.

Slope of Dew Point Lines

The enthalpy of air (total heat) is computed by means of the following equation developed in HPAC, January, 1938, p. 3.

$$h = ct + 1061w \quad \ldots\ldots [19]$$

Differentiating and holding w constant,

$$\frac{dh}{dt} = c \quad \ldots\ldots [20]$$

Obviously for the water-air chart,

$$\frac{h - h_1}{t_t - t_{t1}} = N = \frac{dh}{dt}$$

Hence,

$$N = c \quad \ldots\ldots [21]$$

[This is the fourth of four articles on the water-air chart. The first appeared in the April, 1941, HPAC.]

AIR CONDITIONS FOR EFFICIENCY OF DRAFTSMEN

Despite the fact that the Tennessee Valley Authority recently advertised in the newspapers to ask citizens to conserve in their use of electricity so that ample power will be available for national defense production, air conditioning is regarded as so essential to efficiency in its drafting rooms that it is installing equipment in two of its buildings, it is reported.

The air conditioning is needed, it is understood, because draftsmen are working under pressure, and one sweep of the arm by a perspiring draftsman can spoil the work of several days.

In this connection, it is interesting to recall an article by Paul G. Heidman, superintendent of buildings of the Detroit Edison Co., which was published in the October, 1939, HEATING, PIPING AND AIR CONDITIONING. This article reported a study made in the drafting and surveying bureau of the Detroit Edison Co. which indicated that the efficiency of this bureau has increased very markedly since it has been housed in air conditioned space. It was also found that the total operating and maintenance costs for the new air conditioned service building were 39.7 per cent lower than those in the company's general office building (without air conditioning), labor costs were 40.2 per cent lower, steam costs 62.8 per cent lower and cleaning costs 42.7 per cent lower. Electricity costs —including the power for all air conditioning equipment — were found to be only 1.6 per cent higher, although the light intensity is twice that in the general office building.

GASKETED JOINTS for high pressure high temperature PIPING SERVICE

By Arthur McCutchan

JOINTS employing gaskets range from the lowest to the highest pressures encountered in handling fluids in the liquid, vapor or gaseous state. Extreme ranges of temperature also are encountered. Obviously, the requirements for and characteristics of gasket materials differ greatly for flanged joints used under these widely different conditions. It is equally obvious that the author's experience with gasketed joints does not cover the entire range of either pressure or temperature. This article, therefore, is limited to a discussion of gasketed joints as used in central station power plants and in two experimental high temperature installations.

Joints for 900 Psi, 900 F

While the development of welded joints has resulted in the almost complete elimination of flanged line joints in superheated steam lines, there are occasional joints at equipment where ready disassembly is desirable.

The overall conclusion from experience with flanged joints on 1000 F and 1100 F experimental steam lines is that better chance of success will result if extremely heavy flanges and bolting are used. The need for such heavy construction is not apparent from consideration of internal pressure alone, but may be demonstrated by an analysis of the effect of a bending moment acting on a joint. This moment may be caused by bending moments resulting from constraint of thermal elongation of the line or by water collecting on one side of the joint.

The following analysis of the effect of a bending moment acting on two typical flanged joints may serve to illustrate this point. For purposes of illustration, the following assumptions have been applied in turn to raised face joints made up with 1500 and 900 lb flanges of the American Standard ASA B16e.

1) Service condition of 900 psi, 900 F.

2) 12 in. flanged joint between flanged valve and Schedule 100 pipe.

3) 3/8 in. wide metal gasket located near inner edge of ground contact surface of raised face.

4) Bending moment of 472,500 in.-lb corresponding to a total combined bending and longitudinal pressure stress of 10,000 psi.

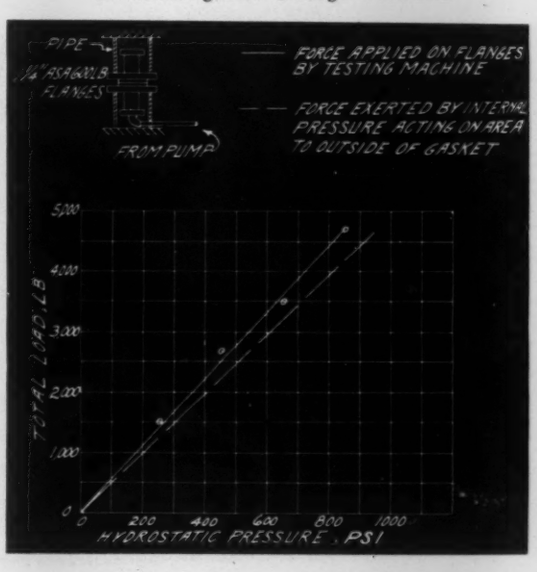
5) Bolting material capable of sustaining a residual stress of 11,000 psi after relaxation.

The effect of this bending moment on the residual compression on the gaskets of 1500 lb and 900 lb flanged joints is shown by comparing the final conditions with and without bending moments acting, Tables 1 and 2.

The results of the comparison in Tables 1 and 2 indicate that the residual compression on the narrow metal gasket is seriously reduced by

GASKETED JOINTS—In this discussion of gasketed joints for high pressure, high temperature service, Mr. McCutchan (engineer, engineering div., the Detroit Edison Co.) points out that various types of gaskets can be used with good success provided flanges and bolting are sufficiently heavy. It is important to estimate the size of bending moments acting on flanged joints, as their magnitude has a pronounced effect on the ability to keep such joints tight. Generally speaking, the larger the bending moment obtaining, the greater the necessity for using heavy flanges for a given service condition. . . . Likewise, the effect of water lying on the bottom of a horizontal joint may have much the same effect as too large a bending moment. . . . The possibility of compressive creep failure when using narrow metal gaskets in high temperature joints also should be recognized

Fig. 1—Test showing load on flanges at which gasket leakage occurred



action of the assumed bending moment in the case of the 900 lb flanged joint. The 1500 lb joint with this same moment acting is able to maintain a pressure on the gasket of better than 19 times the internal pressure of 900 psi as compared with two times for the 900 lb joint. Just how low the gasket compression can be reduced before leakage will occur is considered in the following section, where it will be shown that a residual gasket compression of only two times the internal pressure is about the practicable minimum.

Minimum Bolt Force to Prevent Leakage

The residual compression on a gasket to prevent leakage is dependent upon how effectively the gasket has been caused to adhere to the gasket contact surfaces, which—in turn—is considerably affected by the condition of the contact surfaces and the gasket material and design.

Composition Asbestos Gaskets

The following test (made by Detroit Edison research department, June 16, 1932) was run to secure definite data on the amount of end force or bolt load that must be maintained to prevent leakage of an American 600 lb raised face joint using an asbestos composition gasket between serrated gasket contact surfaces.

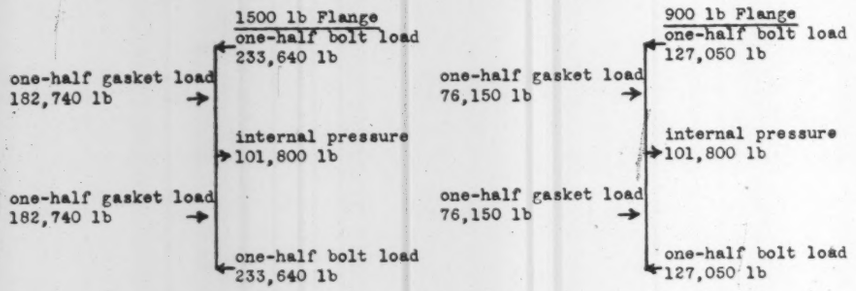
The test setup is illustrated by the sketch on Fig. 1. A new asbestos composition gasket $\frac{1}{16}$ in. thick was inserted between the contact surfaces and the joint placed between the platens of a testing machine. The load was applied to the flanges on the bolt circles by means of short lengths of pipe. The load on the testing machine was first run up to approximately 10,000 lb in order to set the gasket in place. This load corresponded to the bolt loading which would result if the bolts in the 1¼ in. joint were initially tightened to give a gasket compression of six times the internal pressure of 400 psi with the pressure acting on the joint. In this case, the internal pressure of 400 psi was considered as acting over the area to the outside of the gasket contact surface.

The results, as given on Fig. 1, show that a gasketed raised face joint did not leak as long as the force applied on the bolt circle exceeded by approximately 15 per cent the force exerted by the internal pressure, considered as acting over the area bounded by the outer periphery of the gasket contact surface.

It should, of course, be recognized that the loading on the flanges in this test was uniformly distributed around the bolt circle and that no bending moments, temperature differences, or other disturbing influences were present. The results, however, do give an indication of the minimum residual compression for this type of joint under favorable conditions.

As a matter of fact, these tests which were made in 1932 show somewhat better results for this type of joint than were found by the elaborate series of tests reported by F. J. Cowlin and J. P. Chittenden in their paper Joints for Steam Pipe Lines before the [British] Institute of Marine Engineers, February 14, 1939. On the basis of their tests, when considering the internal pressure as acting only on the area given by the inside diameter of the gasket, Messrs. Cowlin and Chittenden con-

Table 1—Final Condition With No Bending Moment Acting



$$\text{Compression on gasket} = \frac{\text{force at center of gravity of gasket semi-circle}}{\text{one-half area of gasket}}$$

1500 lb flange, 900 lb SSP

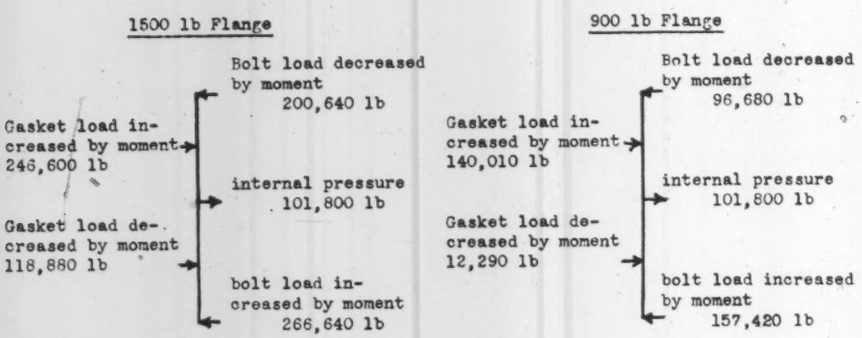
$$\frac{182{,}740}{6.85} = 26{,}677 \text{ lb}$$

900 lb flange, 900 lb SSP

$$\frac{76{,}150}{6.85} = 11{,}116 \text{ lb}$$

Gasket compression in terms of working pressure $= \frac{26{,}677}{900} = 29.64 \times$ SSP for 1500 lb flange and $\frac{11{,}116}{900} = 12.35 \times$ SSP for 900 lb flange. ("SSP" is steam service pressure)

Table 2—Final Condition With Bending Moment



Compression on reduced pressure half of gasket:

1500 lb flange, 900 lb SSP

$$\frac{118{,}880}{6.85} = 17{,}354 \text{ lb}$$

900 lb flange, 900 lb SSP

$$\frac{12{,}290}{6.85} = 1794 \text{ lb}$$

Gasket compression in terms of internal working pressure:

$$\frac{17{,}354}{900} = 19.28 \times \text{SSP}$$

$$\frac{1794}{900} = 1.99 \times \text{SSP}$$

cluded that the axial load should be at least two times the hydraulic or internal pressure load in order to prevent leakage. In other words, the efficiency of the joint, as explained in detail in the following section, was considered to be only 50 per cent. Somewhat better results were reported using coarser serrations than the 32 per inch customarily specified for "phonograph" finish. Reducing the width of the gasket from 1⅛ in. to ⅜ in. likewise showed some improvement.

While these test results caused the authors of the above-mentioned paper to conclude such joints were "far from the best," many thousands of these joints on steam and boiler-feed service in this country attest to their general usefulness.

Serrated Metal Gaskets

Owing to the lack of published information about the behavior of serrated metal gaskets, the following test results reported by Messrs. Cowlin and Chittenden may receive more consideration. A relatively small number of coarse serrations was found to give much higher so-

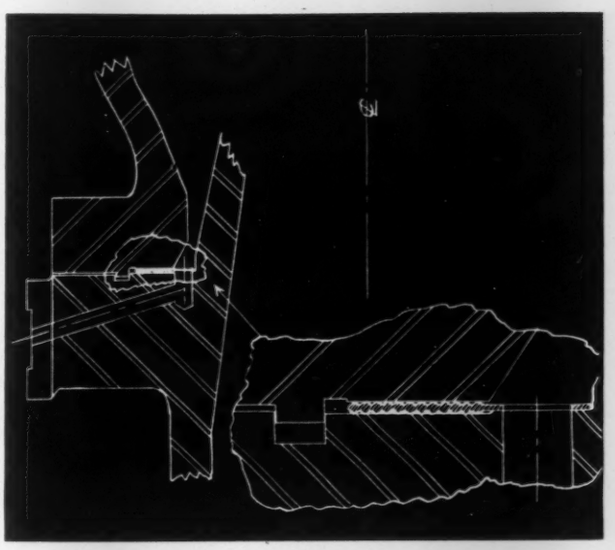


Fig. 3—Detail of bonnet flange showing the re-located serrated metal gasket

cent for those having 12 to 15 serrations. Even higher efficiencies were reported for two serrations but the results were more erratic. The pitch of the serrations on all the gaskets tested was 12 per inch. The recommendation of the authors of the above paper to use a gasket ¾ in. wide with about eight serrations possibly should be limited to the 3 in. pipe size for which the tests were made.

In any case, these tests on serrated metal gaskets would seem to show that a properly proportioned gasket of this type should perform the function of a gasket in conjunction with ground contact surfaces as well or better than composition asbestos gaskets have with serrated

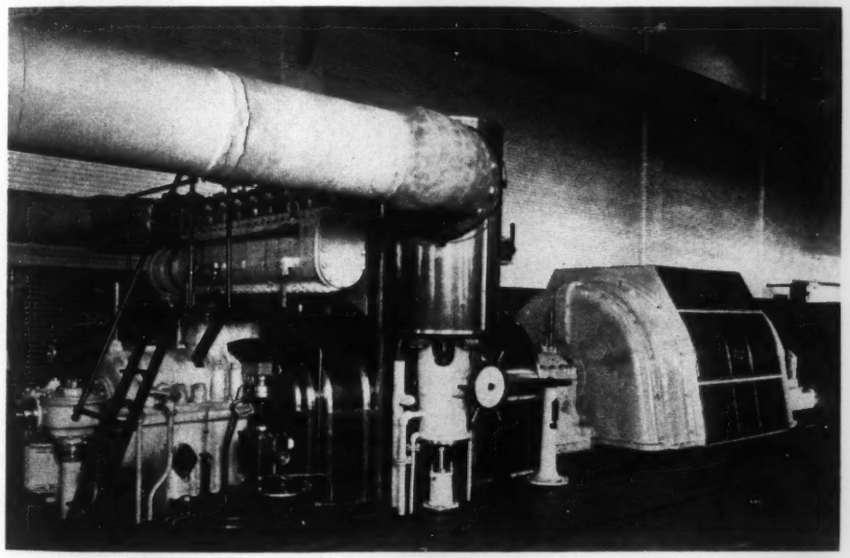


Fig. 2—10,000 kw back pressure turbogenerator

called joint efficiency than a larger number of fine serrations. Joint efficiency was defined as follows:

$$E = \frac{\text{total hydraulic load, lb}}{\text{axial load, lb}}$$

$$= \frac{\text{internal pressure, psi} \times \text{area to inside of gasket, sq in.}}{\text{axial load, lb}}$$

On this basis, efficiencies of 70 to 80 per cent were reported for gaskets having between four and eight serrations as compared with 40 per

contact surfaces in the temperature range for which composition asbestos gaskets are suitable.

Application of a Serrated Metal Gasket to a Modified 900 Lb Flanged Joint

In rebuilding the Delray experimental 10,000 kw, 400 psi, 1000 F turbine for use as a back pressure unit taking steam at 815 psi, 900 F and exhausting into the 400 psi, 700 F plant header, it was found possible to re-use the bonnet and a few other parts of the emergency trip stop valve. A duplicate of this valve was made and the new steam chest arranged to be fed from both ends. One of these valves is shown at the right of the steam chest in Fig. 2.

The flanges on the new steam chest were thickened up to approximate the American 1500 lb standard, but the mating flanges on the valve bonnets were left approximately equal to the 900 lb standard. The resulting flanges, which are 5 in. and 3 in. thick, respectively, are shown in Fig. 3.

In order to minimize the duty on the flanges and bolting, the turbine manufacturer installed an ingenious design of gas filled gasket. (A description of gas filled gaskets was given in the October 8, 1937 issue of the [British] Engineer, p. 385. Also see p. 568, HPAC, September, 1939).

The gas filled tubular gaskets were located in the recesses between the bottom of the tongue and the groove (see Fig. 3). While this type of gasket, if properly proportioned and installed so as to prevent crushing, would seem to have interesting possibilities, its development had not progressed sufficiently at the time of this installation with the result that the gas filled gaskets were ruined and the joints had to be redesigned for the more usual type of gasket.

The simplest procedure would have been to install a solid gasket in

the space provided for the gas filled rings. The area of the resulting gasket would, however, have been too small to sustain the heavy bolting and to utilize the heavy flanges to their fullest advantage.

If a metal gasket of appreciable thickness is to be used under temperature conditions producing compressive creep in either the gasket material or in the gasket contact surfaces, the width of the body of the gasket should be correctly proportioned to the bolt area.

One of the causes of joint leakage encountered in the 1000 F steam lead to the experimental 10,000 kw turbine was deformation of the flange faces by compressive creep. (See High Temperature Steam Experience at Detroit, by R. M. Van Duzer, Jr. and Arthur McCutchan, ASME Transactions.) The indentation of the flange face by the relatively narrow metal gasket, 0.39 in. wide, is shown in Fig. 4.

In order to obtain a pressure seal with a metal gasket it is necessary either to have perfect ground or lapped surfaces on the joint faces or to apply sufficient unit pressure on the gasket to cause it to flow into and fill the imperfections.

On low temperature services this necessity of securing high unit compression on a metal gasket does not preclude the use of narrow plain gaskets, but where creep is a controlling factor it is important to provide a cross-sectional area in the body of the gasket which will be comparable in strength to the long time strength of the bolting and flange material. The use of a coarse serrated metal gasket enables the necessary high unit compression to be obtained on the gasket contact surfaces to effect a pressure seal while at the same time the compressive stresses in the body of the gasket can be kept within reason.

In the case of the trip stop valve bonnet joints, these conditions were fulfilled by re-locating the gasket as shown in Fig. 3. It was necessary to machine and lap the new contact surfaces on the bonnets, but those on the valve bodies required only lapping. Plates for lapping and surface plates for checking the finished surfaces were machined from 14 in., 250 lb screwed "ferro-steel" pipe flanges obtained in the undrilled condition.

A ⅛ in. thick serrated monel gasket having eight serrations per inch and an overall contact width of 1½ in. was provided. The total bolt root area of the twenty 1¾ in. bolts was 39.6 sq in. The area of the body of the gasket was 89.5 sq in., while the area of the serrations, contacting the ground surfaces, assuming them flattened to give a width of 0.04 in., was 31.25 sq in. With an initial bolt stress of 40,000 psi this gives an initial stress condition of 17,700 psi in the body of the gasket and 50,700 psi on the serrations.

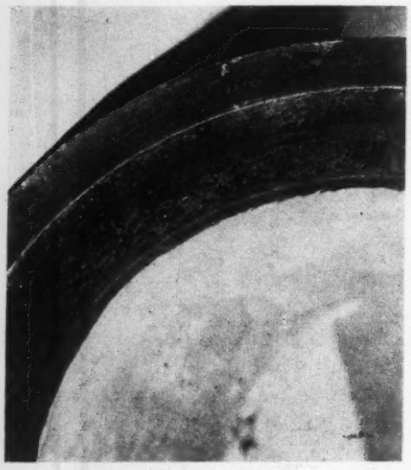


Fig. 4—Indentation caused by narrow metal gasket

For a residual stress of 11,000 psi in the bolting, the residual compression on the serrations with the internal pressure of 815 psi acting is 6.5 times the working pressure.

The bending moments acting on these joints from thermal expansion of the steam line are relatively insignificant so the only cause for uncertainty as to joint tightness arises from the possibility that water may collect on the joint during starting up periods. If the drains, one of which is shown on Fig. 3, are left open until the line is thoroughly warmed up, leakage should not occur. Some seepage has been reported during the starting up periods, but the joints have otherwise been satisfactory.

Effect of Water on One Side of Steam Joint

Wherever water can collect in sizable quantities on one side of a steam joint during starting up periods or on a dead end line (as in the case of a valve bonnet joint installed with the stem horizontal), possible joint troubles can be anticipated. The chances of trouble are less, however, if the residual gasket compression is kept high through the use of heavy flanges and bolting and correct design of gasket.

The most obvious way of eliminating trouble where water may collect on a bonnet joint or other flanged joints in steam lines would be to install drains, but plant operators always are opposed to the installation of any additional drains, traps, etc.

The following instances of joint difficulties caused by differences in temperature between the top and bottom of the line are cited to show that trouble from this cause can be anticipated.

Leaks in Low Pressure Condensate Lines

Leaks at cast iron screwed flanged joints were attributed to the observed fact that the bottom of the line was colder than the top. This difference in temperature was caused by an accumulation of water in the bottom of the line, while the top was maintained at a higher temperature by the condensing vapor. This difference in temperature induced a bending moment which attempted to pry the joints apart. In several cases the back of the screwed flange hub was found to be stretched to such an extent by this prying action that the flange was loosened from the pipe. Replacement of these cast iron flanges by steel flanges made it possible to obtain sufficient increase in gasket compression to counteract this tendency for a difference in temperature between the top and bottom of the line to pry the joint apart.

Breakage of Welded Heating Header

In the days of bare wire welding, a welded heating header broke during the initial warming up of the line. This breakage was attributed to a temperature differential between the top and bottom of the header caused by a very slow admission of steam which permitted the steam to drift along the top of the line. Steam in contact with the upper surface of the pipe condensed, giving up its latent heat to the metal. The condensate, containing only its sensible heat, then collected in the bottom of the line and was drained

out in an undercooled condition from contact with the cold side of the pipe. In this case, where a volume of steam insufficient to fill the pipe was admitted, almost five times more heat was available to heat the top than the bottom of the line. A more rapid warming up of this header has prevented a recurrence of the original failure.

Leakage of Water at Spring Compensated Joints in 1000 *F Steam Line*

Leakage of water at the spring compensated joints in the 1000 F steam line to the experimental 10,000 kw turbine at Delray during the starting up periods was explained as caused by using a concentric reducer between the 10 in. valve at the superheater outlet and the 8 in. steam line. A puddle of water collected on these joints during warming up of the line and again reluctance to provide suitable drains in combination with lightly loaded joints caused leakage. Since these joints were steam tight in service, water seepage during starting up was regarded more as a minor annoyance than an operating disability.

Water Accumulation Cause of Trouble

Because gaskets and joint design often are blamed where water accumulation is the real cause of joint trouble, the following explanation of the manner in which a temperature difference between the top and bottom of a joint may produce a bending moment sufficient to pry a joint apart seems worth reproducing here.

Analogy with Heated Bar—Consider a metal bar simply supported as shown in *A* of Fig. 5. If this bar is heated by a flame from the top as indicated in *B*, the bar will expand more on top than on the bottom. The center line of the expanded bar will approximate an arc of a circle. The center line of a beam bent by couples acting at the ends, *C*, also is considered as an arc of a circle.

The similarity between the curvature produced by heating one side of a bar more than the other and the curvature induced by bending couples acting on the ends enables a relation to be determined between the temperature difference of the two surfaces and the couples necessary to prevent rotation of the ends of the bar.

The magnitude of the end couples for perfect fixity, *D* of Fig. 5, is just sufficient to annihilate the curvature produced by the difference in expansion of the top and bottom of the bar. The following equation [See Strength of Materials, by S. Timoshenko, Part I, p. 100. (D. Van Nostrand Co., 1930)] is based on this condition:

$$M = \frac{EI\,a\,(t_1 - t_0)}{h}$$

where a = average coefficient of expansion for the temperature range t_0 to t_1 = inches per inch per degree Fahrenheit;
h = depth of bar or pipe diameter, inches;
I = moment of inertia of bar or pipe, inches to the fourth power;
E = modulus of linear elasticity, pounds per square inch.

Application of Equation for Bending Moment Caused by Temperature Difference to Flanged Joint—In the case of the 10 in. valve joint in the 1000 F steam line the following values for a section of the cylindrical part of the valve body immediately adjacent to the flange may be substituted in this equation: $E_{400F} = 28{,}000{,}000$ psi; $I = 755$ in.^{4}; $a = 0.0000065$ in. per in. per F; $h = 12\frac{1}{2}$ in.

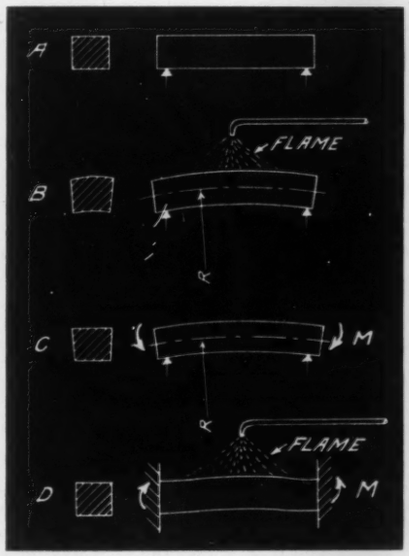


Fig. 5—Relation between expansion of bar heated on one side and restraining moment

$$M = \frac{28{,}000{,}000 \times 755 \times 0.0000065\,(t_1 - t_0)}{12\frac{1}{2}}$$

= 11,000 in.-lb per F difference between top and bottom of pipe.

For an initial gasket compression of 10 times the steam service pressure, a difference in temperature between top and bottom of pipe or valve body adjacent to the joint of only 290,000 ÷ 11,000, or 26 F, apparently would cause leakage. The figure of 290,000 in.-lb was determined as the bending moment necessary to reduce the gasket compression at the bottom of the joint to zero, using the same method of analyzing the effect of a bending moment on this joint as was followed in the preceding section entitled Joints for 900 Psi, 900 F.

The question might be raised as to why flanged joints ordinarily are able to withstand temperature differences of this kind. An important factor—if not the whole answer—is the gasket compression which the flanges and bolting are capable of maintaining. For example, a 600 lb ASA flanged joint for 400 lb SSP as customarily set up has an initial gasket compression of 30 to 40 times the SSP, while the spring compensated flanges which leaked water were adjusted to give an initial gasket compression of only 10 times the SSP because the bolt loading was limited by possible stress in the hairpin spring. These relations simply mean that it would require a temperature difference of three to four times the magnitude of that just determined to pry the customary joint apart.

Experience With Ring Joint

Experience with a ring joint on the trip stop valve bonnet joint of the 10,000 kw, 1000 F turbogenerator was not entirely satisfactory as the joint had to be remade after a years' service. No difference was found in the performance of the soft iron ring first installed and the monel replacement ring. Both rings were of the oval type rather than the octagonal design now advocated for steam service. A 5 in., 400 lb carbon moly ring joint in the 380 psi, 925 F section of the experimental steam line stayed tight until disassembled by mistake after some 12,000 hr service. It was reassembled with the same ring and bolting and still is under test.

News about new and improved equipment appears in the department "Equipment Developments" in HPAC each month.

INCREASED TAXES AND SELLING COSTS MAKE EFFICIENT OPERATION IN PLANTS AND BUILDINGS MUCH MORE IMPORTANT

A. J. RUMMEL REPORTS ON SURVEY OF AIR CONDITIONING OPERATION, SHOWS HOW AN ANALYSIS AND THE PROPER INSTRUMENTS CAN REDUCE THE COST OF POWER NEEDED

EFFICIENT OPERATION—Increased efficiency in the operation of mechanical equipment serving a plant or building—such as the air conditioning system—reflects itself in net profits just as much as does efficient operation of other departments of the business. With increased taxation and selling costs squeezing out profits to the point where an increase in sales no longer results in a normal increase in profits, improved operating efficiency is more important than ever before. . . . A two year study on how large air conditioning systems are actually operated was recently completed, and numerous opportunities for savings were revealed. For instance, it was found that in many cases the compressors were run at times when they were not actually needed. Mr. Rummel, air conditioning engineer with the San Antonio Public Service Co., analyzes a typical case here, and shows how the electric power bill for one month was $161.62 more than was necessary. While the case he describes is that of a department store, the same principles apply to other types of business establishments as well

IN an effort to determine exactly how air conditioning systems are being operated and the extent to which economies in operation can be made, a two year detailed study of numerous installations has just been completed. This study of operating records and methods of large air conditioning systems in different types of business establishments in the Southwest revealed many inefficient operating practices. It also brought out the fact that in most cases, if thorough and complete analyses of factual data pertaining to operation were made, the inefficient practices could be determined and remedied and considerable savings in operation costs could be realized.

Space does not permit a complete discussion of all the factors noted in the various types of businesses investigated, but one of the more interesting groups studied was the department store. Before discussing in detail the operating savings that could have been realized by one particular store if proper factual studies had been made of the method employed in operating the air conditioning system, it will be well to analyze briefly the more or less generally found operating policies adopted by most department store managements.

Operating Efficiency Getting More Attention

Most executives feel that although department stores are at present enjoying a period of rising sales, increased taxation and selling costs are squeezing out profits to the point where an increase in sales no longer results in a normal increase in profits. For this reason improved operating efficiency in all departments is receiving increased attention. This is probably typical of all fields of business.

In most cases, department store operation expense is classed under two general headings: first, the cost of operating that part of the business directly related to the purchase, advertising and sale of merchandise; and second, the cost of that part indirectly connected with the operation of the business, such as the operation and maintenance of the mechanical equipment.

In the various operating departments directly connected with the sales end of the business are found men who, in most cases, are well qualified and trained to operate efficiently their particular part of the business. Regardless of the already established competency of these men, most managements consider it good business to have them constantly on the alert for ways and means of further increasing the efficiency of operation of their respective departments. Sales meetings; local, district and national conferences; trade magazines—all are considered essential. As increased

"Buried treasure" is found in increasing efficiency of operation of mechanical equipment in plants and buildings, for it reflects itself in net profits just as much as does the efficient operation of other departments of business establishments

efficiency in operation of the mechanical equipment reflects itself in net profits just as much as does the efficient operation of the other departments, this same alertness to ways and means of improving operation is just as important in this department as it is in the others.

When the monthly power bill for air conditioning is received by store managements it is usually given a casual inspection and considered satisfactory if the amount of the bill is about the same as that for the preceding month, or for the same month the preceding year. If instead of this casual examination of the bill, a careful study and analysis were made of all factors that entered into the total, inefficiencies in operation might be indicated.

Cost of Inefficient Operation

In order to study the method of operation of the various air conditioning systems and the resulting operating costs, separate recording demand and kilowatthour meters were installed in the electric circuits feeding the air conditioning equipment. This metering arrangement was continued for a period of two years and a study of these records disclosed many facts relative to the various methods of operation employed and considered correct. The following is an interesting example indicating what inefficient operation actually costs.

In the Southwest, there are periods during the fall and winter months when some cooling is necessary in department stores due to high outdoor temperature conditions—especially where high internal heat loads exist. Although it is often necessary to operate the system at times during these months, it was found that the majority of the stores studied often operated their compressors when it was absolutely unnecessary. In other words, by taking into consideration all factors, the same indoor temperature conditions could have been maintained without the operation of the compressors as were maintained by operating the compressors. By not operating the compressors, the increase in demand established by the compressors would be eliminated, resulting in a lower monthly power bill.

Although most store managements say that they do not operate their equipment until the temperature reaches a point where operation is necessary, the following explanation will show how the same indoor conditions can often be maintained without compressor operation even though the dry bulb temperature has reached the point where operation of the compressor is deemed necessary.

Let us consider a typical department store similar to the ones surveyed. This typical installation consists essentially of an electric motor driven compressor or compressors and a water cooler, situated in the basement of the store. Chilled water from the water cooler is pumped to a number of spray type coolers and dehumidifiers at strategic points in the building, each spray unit being provided with an outside air connection. The average system of this kind is usually designed and installed so that the amount of air needed properly to cool the building leaves the spray dehumidifier and cooler approximately saturated at somewhere around 58 F.

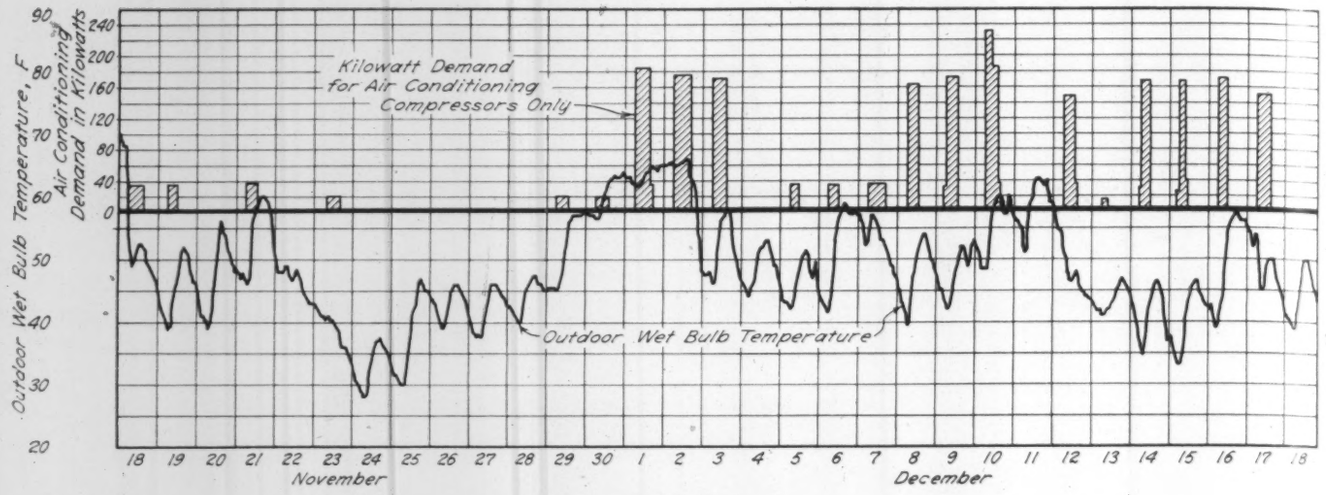
On days of rather high outside dry bulb temperature during the fall and winter months, and especially on days when the store has a high internal heat load (as for instance in the case of sales), the operator is instructed to start the air conditioning system. Although the outdoor dry bulb temperature might be 80 F or above, often the outdoor wet bulb temperature is below 58 F. If, when the operator is requested to start the air conditioning system, he would determine the outside wet bulb temperature, he could determine whether it was necessary to start the compressors or, if the outside wet bulb temperature was below 58 F, if all outside air should be circulated through the sprays. If the recirculated air is closed off and outside air at a wet bulb temperature of 58 F or less is circulated through the sprays, it is apparent that the air leaving the sprays will be at approximately the wet bulb temperature of the outside air and at the same temperature that is maintained with the compressors operating and the air recirculated.

Despite rising sales, increased taxation and selling costs squeeze out profits, make increased operating efficiency all the more important

Surprising Savings

Using the typical store above described and applying actual records of outdoor wet bulb temperatures obtained by means of a recording wet bulb thermometer and recorded compressor operation, surprising savings that could be realized by attention to this one criterion of operation are revealed.

Due to diversity of load the kilowatt demand of this store for lighting, elevator and miscellaneous power use varies from 342 to 392. The added load for the operation of the various fans and pumps for distributing the cooled, dehumidified air in summer and warm, humidified air in winter is approxi-



Unnecessary compressor operation when the outdoor wet bulb was under 58 F cost $161.62 in one month

mately 28 kw, making a total demand of somewhere between 370 and 420 kw for what might be considered normal operation when the compressors are not in use.

The particular power rate in accordance with which this customer is billed is on the basis of an energy charge based on certain hours use of the billing demand. The monthly demand for billing purposes is the average of the highest 15 min demands recorded during each of six periods during the month as follows: First period, 29th day of one month to the 3rd day of the following month, inclusive; second period, 4th day to the 8th day of each month, inclusive; third period, 9th day to the 13th day of each month, inclusive; fourth period, 14th day to the 18th day of each month, inclusive; fifth period, 19th day to the 23rd day of each month, inclusive; and sixth period, 24th day to the 28th day, inclusive.

An examination of one month's demand chart for the total store load showed that the highest demands recorded for the six billing periods were 406, 397, 588, 562, 622 and 579—resulting in an average of 526, which was used as the basis for billing purposes for this month.

The corresponding demand chart of the air conditioning system was studied for this same period and the demands were plotted on a graph which also showed the outside wet bulb temperature. It was noted that the compressors were often operated during periods when the outdoor wet bulb temperature was below 58 F. In other words, if all outside air had been passed through the sprays and circulated through the building, the same indoor temperatures would have resulted as did with the compressors operating and the greater portion of the air recirculated.

The operation that would have given the same indoor results at a lower cost is noted in Table 1.

In addition to establishing an increase in average demand of 94 kw (526—432) that was not necessary, 11,270 kwhr was used while these compressors were unnecessarily operated. This increase of the demand 94 kw and the 11,270 kwhr consumption resulted in the electric power bill being increased $161.62, which amount could have been saved by a close and intelligent study of operation. The same analysis for other months revealed similar savings that could have been made.

After the outdoor wet bulb temperature reaches 58 F and above, and compressor operation is required, additional savings can be realized by proper attention to indoor and outdoor conditions. In most cases when the compressor and entire system was in operation, it was found that the amount of recirculated air and outside air used was never changed—regardless of the outside conditions.

For instance, in the above example when the compressors are on and the system operating, the air temperature leaving the sprays is approximately 58 F. The air returning from the store is at about 79 F and 50 per cent relative humidity, which corresponds to a wet bulb temperature of 66 F. Therefore, since the wet bulb temperature is an indication of the total heat, it can be readily seen that even if the outdoor dry bulb temperature is above 79 F, if the outdoor wet bulb temperature is between 58 F and 66 F the system would be operating at reduced load if the recirculated air were cut off and all outside air passed through the sprays, circulated through the building and discharged to the outside.

Table 1—Analysis of Demand Actually Established and Demand That Would Be Required With More Careful Operation

Billing Period	Kw Demand Actually Established	Kw Demand Required	Comments
19th to 23rd.....	406	406	Normal operation
24th to 28th.....	397	397	Normal operation
29th to 3rd.....	588	588	Compressor operation necessary on 1st and 2nd but not on 3rd. Demand that was established justified but not kwhr consumption on 3rd
4th to 8th.....	562	400	Compressor operation not necessary during this period. Operation of compressor for one day, the 8th, established demand for period
9th to 13th.....	622	400	Compressor operation not necessary during this period
14th to 18th.....	579	400	Compressor operation not necessary during this period
Average monthly demand.....	526	432	

Note: Demand for equipment other than air conditioning compressors varies from 370 to 420 kw, with 400 considered normal.

Construction Details for
RADIANT HEATING AND COOLING

J. Napier Adlam Describes Installation at Northwestern Technological Institute

THE special radiant heat panels installed in one of the large classrooms at the new Northwestern Technological Institute, Evanston, Ill., where Prof. B. H. Jennings hopes to obtain some very valuable information on both radiant heating and cooling, should be of considerable interest to architects and heating engineers. The total panel surface provided is sufficient to allow several panels to be cut out as desired, so that the relative effectiveness of the different locations may be studied. A large portion of the floor area is to be heated with three wrought iron pipe coils embedded in the concrete, while six large metal panels will provide heat on the side walls and ceiling.

Prof. Jennings has, during the last four or five years, carried out a number of experiments in this field at Lehigh University, Bethlehem, Pa., and with these added facilities now at his disposal he hopes to obtain some very practical information which will

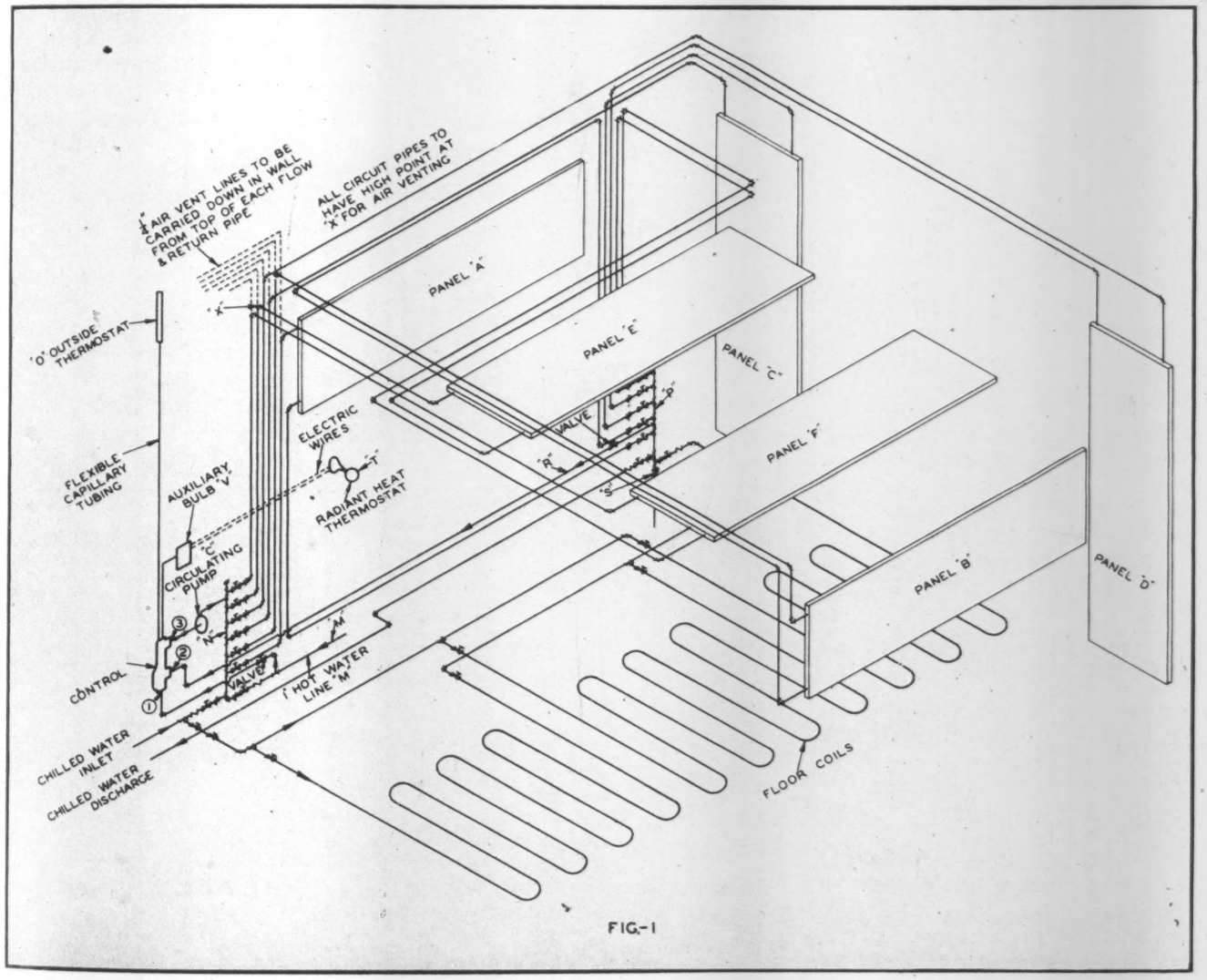


FIG.-1

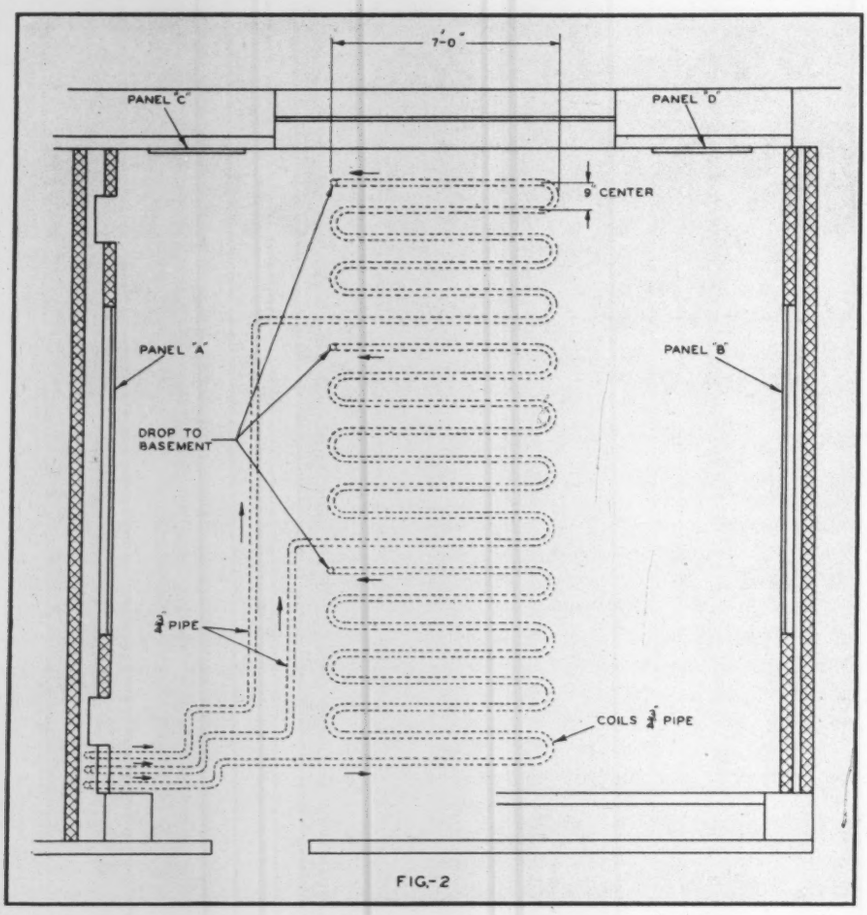


FIG.-2

be useful to the trade. One of his main objects will be to discover the difference, if any, in the efficiency of panels according to their positions. He intends also to carry out a series of tests to determine the value of radiant cooling and to discover how far this can be extended to promote comfort without meeting difficulties with moisture. It has already been established that it is possible to reduce the refrigeration load in summer time by providing cooled surfaces to increase the radiation loss from the human body, and it is hoped that some reliable data will be forthcoming from Northwestern Tech.

Fig. 1 illustrates the general layout of the panels and also the connecting pipes from the main hot water heating circulation. Panels *A* and *B* are placed opposite each other at high level on the two side walls, respectively; panels *C* and *D* are both placed on one end wall in a vertical position; while panels *E* and *F* are placed on the ceiling. Isolating valves are installed on both the flow and return circuits to each panel so that all or any of the panels may be shut off as desired. Connections are also provided to circulate chilled water through the coils.

An automatic control is provided to maintain the correct water temperature circulating through the panel coils, according to the outside weather conditions.

Fig. 2 illustrates the three ¾ in. wrought iron pipe coils embedded in the concrete floor, and Fig. 3 shows the method of supporting the coils in the floor construction. It was thought desirable to provide more positive venting in Fig. 2, so during installation the outlet points instead of dropping directly to the basement were run over to the wall partition behind panel *A* and vents were supplied for use if needed.

Fig. 4 gives constructional details of panels *A* and *B*, which are installed lengthwise on the side walls at high level. The pipe coils are made from ¾ in. wrought iron pipe and the pipes are spot welded at 4 in. intervals to the No. 11 ga, 0.125 in. steel plate. Tapped bosses are welded

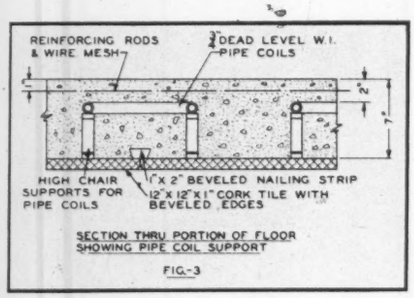

REINFORCING RODS & WIRE MESH
¾" DEAD LEVEL W.I. PIPE COILS
HIGH CHAIR SUPPORTS FOR PIPE COILS
1" X 2" BEVELED NAILING STRIP
12" X 12" X 1" CORK TILE WITH BEVELED EDGES
SECTION THRU PORTION OF FLOOR SHOWING PIPE COIL SUPPORT

FIG.-3

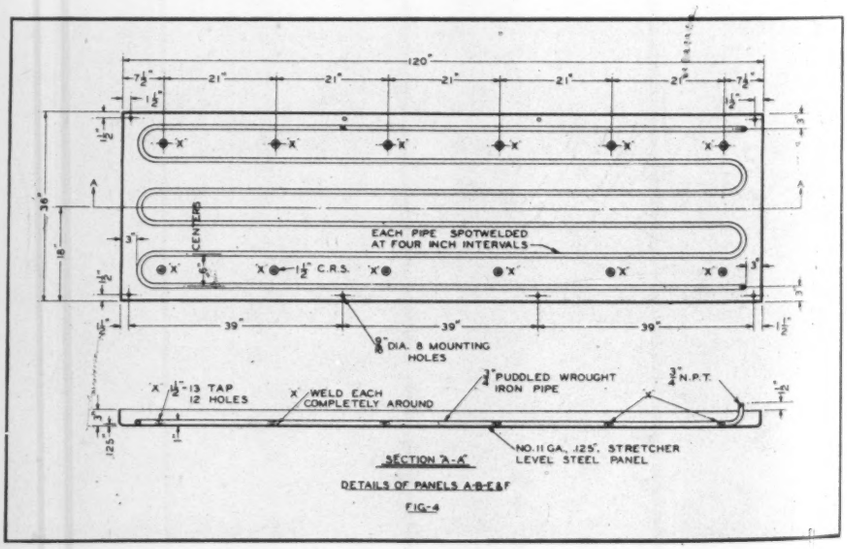

120"
EACH PIPE SPOTWELDED AT FOUR INCH INTERVALS
1½" C.R.S.
⅝" DIA. 8 MOUNTING HOLES
½"-13 TAP 12 HOLES
WELD EACH COMPLETELY AROUND
¾" PUDDLED WROUGHT IRON PIPE
¾" N.P.T.
SECTION "A-A"
NO. 11 GA. .125" STRETCHER LEVEL STEEL PANEL
DETAILS OF PANELS A-B-E & F

FIG-4

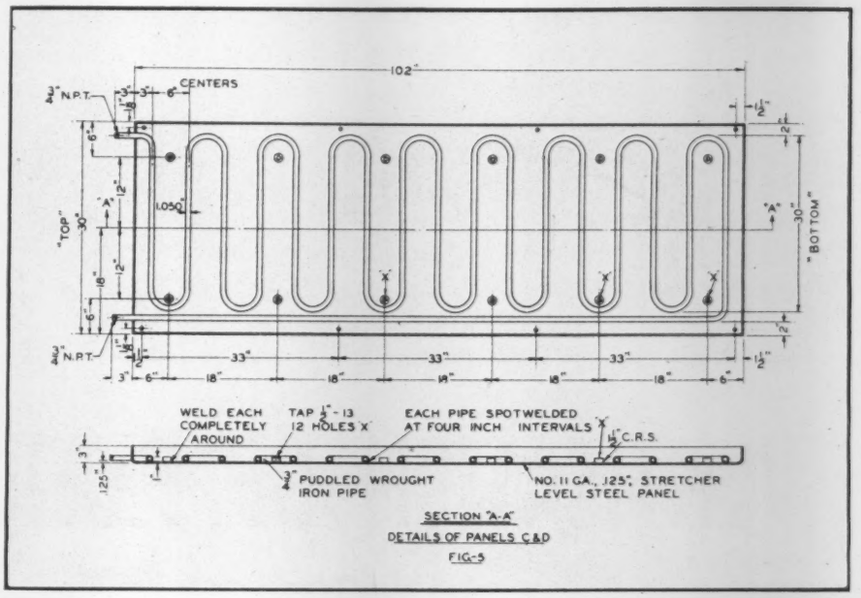
RADIANT HEATING—a subject which almost seems to require an "all-out" effort to satisfy the demand of heating engineers for data and information—is to be studied at Northwestern Technological Institute's new building. Mr. Adlam, vice-president of the Sarco Mfg. Corp., describes the special panels installed in one of the large classrooms there, from which it is expected valuable and practical data on this method of heating and cooling will be obtained. . . . Details of the panel construction are well illustrated in the series of drawings. An automatic control valve will provide the proper temperature of water circulated through the coils, in accordance with the weather

to the plate at points X to make provision for attaching the insulating pads.

The steel panels were fabricated from special stretcher steel plates with return ends formed to give a neat appearance. The surfaces of the iron panels are perfectly flat and will be painted to harmonize with the decorations of the room. Eventually, however, it is planned to make other experiments to ascertain the effect of covering the metal surface with a thin veneer of wood and other material. It will be noted that the pipes for these panels are placed in a horizontal direction to give free access for air venting at high level.

Figs. 6 and 7 show the method of mounting the insulation pads to the back of the panels to reduce the transfer of heat to the walls or ceilings. With panels C and D the circulating pipes are installed on the surface of the walls, whereas the pipe coils for the other panels are connected through the walls and ceiling respectively.

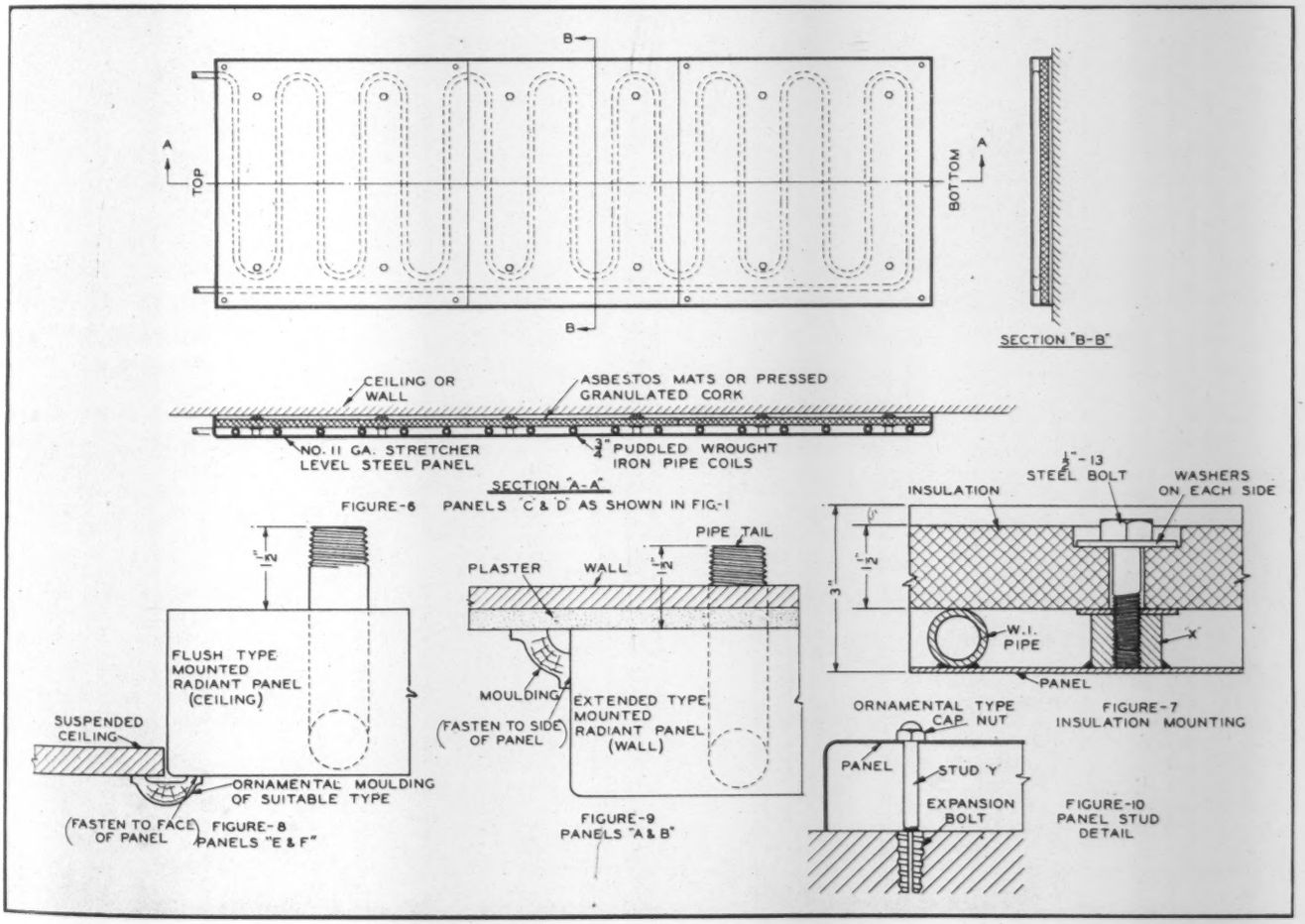
Fig. 8 shows the method of placing a wood moulding over the space between the plaster of ceiling and the panels E and F to give a neat finish and prevent darkening of the surface. Sufficient space is provided to allow expansion of the metal panels without damage to the plaster; and the wood moulding which is fastened to the face of the panel slides over the surface of the plaster quite freely.



SECTION "A-A"
DETAILS OF PANELS C&D
FIG-5

Fig. 9 shows the method of applying a moulding around the joint where the metal panel makes contact with the surface of the plaster on the walls. In this instance the wood moulding is fastened to the side of metal panel, so that the moulding may slide easily over the surface of plaster.

Fig. 10 illustrates the method used for securing the panels to the walls and the ceiling.



Production for National Defense

VENTILATION OF PLATING TANKS

By Allen D. Brandt*

SINCE 1928 when Bloomfield and Blum [Reference No. 1] made a study of the health hazards of chromium plating, considerable attention has been focused upon the ventilation of plating tanks of all kinds. While the recommendations for control of the health hazard by means of transverse or lateral exhaust ventilation given by Bloomfield and Blum are valid even today, several attempts have been made since then to define more specifically the required ventilation rate.

Riley and Goldman [Reference No. 2] in 1937 undertook a short study of plating tanks and presented a simple equation for the computation of the required amount of ventilation (see Table 1). The equation they suggested provides essentially the same ventilation rate as recommended by Bloomfield and Blum. This equation with but minor modifications is in wide use today and is incorporated in the appendix of the proposed American standard safety code for exhaust systems in electroplating operations. More recently Battista, Hatch and Greenburg [Reference No. 3] reported the results of a study on the ventilation characteristics of a plating tank provided with lateral exhaust ventilation. Their results indicate that the quantity of air which must be exhausted for a given velocity over the remote part of the tank is a function of the tank width to the 0.61 power. Silverman [Reference No. 4] in a paper just recently published shows that for a given velocity over the remote corners of a tank, Q (quantity of ventilation per unit time) is a function of the tank length to the 0.85 power and the tank width to the 1.15 power.

Before the results of the studies of Battista, Hatch, Greenburg and Silverman became known to us, the differences in opinion voiced by those people who submitted comments on the proposed electroplating code made it apparent that no uniformity of opinion on the proper ventilation rate or the determination thereof existed among those interested. Consequently, a study was undertaken to determine (1) the air flow characteristics of plating tanks provided with lateral exhaust ventilation and (2) the minimum

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PLATING TANKS—It is essential in connection with some of the electrolytic processes now used in many industries to provide adequate means of protection for the health of the worker. The danger lies chiefly in the exposure to irritating and toxic mists, gases and vapors dispersed from the electrolytic solutions. The design of ventilating systems which will most effectively control these atmospheric contaminants is a practical problem for factory management, one which assumes added significance with the ever-increasing use of electroplating and anodyzing techniques in the manufacture of precision parts and machinery in modern industry. Several of these electrochemical deposition processes comprise important present-day procedures in the production of aircraft, airplane engines and other equipment vital to our defense needs. . . . . Conclusions drawn from this article are as follows: (1) Since the data presented herein essentially substantiate that presented by Battista, Hatch and Greenburg [Reference No. 3], it seems obvious that the center line velocity across a plating tank ventilated by means of lateral exhaust ventilation varies inversely as the distance from the slot to an exponent less than one. That is, the ventilation rate required to maintain a given center line velocity over the tank at the point remote from the slot or slots is given by the equation $Q = KVLW^a$ in which the value of "a" is less than unity. . . . . (2) Inasmuch as the value of the exponent "a" determined by Battista, Hatch and Greenburg is 0.61, in which case the tank had no wall opposite the exhaust slot, and the value of the exponent "a" determined in this study is about 0.5, it appears that the equation $Q = KVLW^{0.5}$ defines the ventilation characteristics of plating tanks more accurately than equations now in use. However, since the value of K is unknown and since the equation $Q = 120LW$, first, has been used widely with good results; and, second, gives results essentially the same as the equation $Q = KVLW^{0.5}$ for a critical velocity V of 75 fpm and for tanks of sizes commonly encountered, it is recommended that it be used pending the determination of the value of K and possible additional data on the characteristics of tanks with two hoods. . . . (3) Since the value of "a" is less than one, it would appear that the ventilation rate per unit tank width decreases as the tank width increases; in other words, the wider tank is favored. However, this condition will exist only so long as the ratio of the tank length to width is such that the velocity contours in the center line plane passing through the slot as an axis are essentially flat for the greater part of the tank length.

Table 1—Equations in Use or Suggested for the Determination of the Ventilation Rate for Plating Tanks

Equation	Source	Remarks
Slot velocity of 2000 fpm and increasing slot width as tank width increases	Bloomfield and Blum	Adequate control but not defined by equation
$Q = 100LW$	Riley and Goldman	Ventilation rate possibly slightly low for adequate control
$Q = 120LW$	American Standards Association. (Outgrowth of equation by Riley and Goldman)	Found to be adequate generally and in very wide use
$Q = KLW^{0.61}$	Battista, Hatch and Greenburg	Value of K not yet determined
$Q = 2.3L^{0.85}W^{1.15}V$	Silverman	

L = tank length, ft. W = tank width, ft. V = velocity, fpm. K = a constant. Q = ventilation rate, cfm.

velocity at the remote point or area over the plating tank necessary to maintain the concentration of chromic acid mist in the air of the workroom and in close proximity to the tank below the permissible safe limit of one milligram per 10 cubic meters of air. Only the first part of this study has been completed and it is to a discussion of the results of this study that the remainder of this paper will be devoted.

Method of Test

A tank 6 ft long by 2 ft wide and with a flat surface about 7 in. below the bottom of the exhaust slot opening was constructed essentially of insulation board. Exhaust hoods having inside dimensions of 6 in. x 10 in. x 6 ft long were constructed of ⅞ in. board. These hoods were provided with an adjustable lip or flange of sheet metal so that the slot width could be adjusted from 0 to 3 in. The lip or flange forming the lower part of the slot extended into the hood proper a distance of 2½ in. simulating to a certain extent the upper edge of a typical plating tank (see section in Fig. 2). Each hood was connected to a 12 in. square exhaust manifold by means of three pipes of 6 in. diameter. The purpose of this layout was to obtain a fair degree of uniformity of slot velocity throughout the entire hood length. The exhaust hoods served as the sides of the tank and in that part of the study devoted to the characteristics of a single hood the unventilated hood was lowered so that its top was at the same level as the lower edge of the exhaust slot.

The velocity determinations were made in a plane normal to the slot and halfway between the ends of the tank. The thermo anemometer was used for the determination of the velocities and several results in the high velocity range were checked with a point velocity type pitot tube [Reference No. 5]. The quantity of air exhausted was measured by orifice meters in the exhaust ducts.

It was intended that a ventilation rate of 120 cfm per sq ft of tank area, as set forth in the proposed American standard safety code for exhaust systems in electroplating operations, would be used. However, the exhauster available for the study moved only 1320 cfm or 110 cfm per sq ft of tank area. For a slot width of exactly 1 in. the slot velocity at this rate of flow would be 2640 fpm. However, with our slot arrangement the width could not be regulated with any great degree of accuracy and happened to be somewhat greater than 1 in., resulting in a slot velocity of about 2200 fpm.

Ventilation Characteristics

A simple mathematical analysis of the problem will demonstrate that for a line source of exhaust of infinite length, V must vary inversely as the first power of X (the distance from the line source) since the successive velocity contour surfaces will be concentric cylindrical surfaces with the line source of exhaust as the axis. Carrying this analysis a step farther, it is obvious that for all practical purposes the same relationship exists in the plane normal to a line source of suction at its midpoint if the source of suction is merely long enough to produce flat velocity contours in a plane through the axis of the line source of suction. Therefore, it seems likely that this relationship, namely, that V varies inversely as the first power of X, would not be modified significantly in the case of a 1 in. slot 6 ft in length so long as no surfaces exist nearby to alter the contour pattern. In other words, the relation between rate of ventilation and tank width is $Q = KLVX$, in which Q = air volume per unit time, K = a constant, L = slot length, V

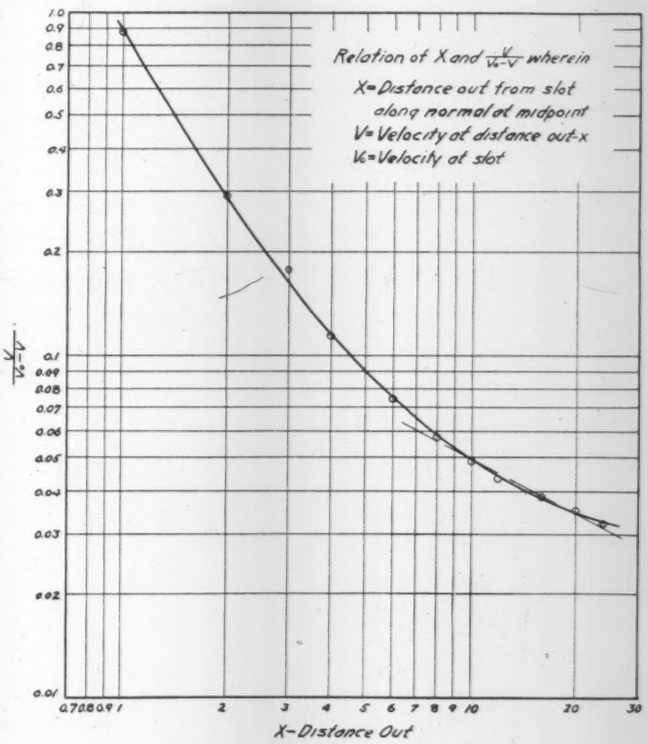

Fig. 1—Relation between center line velocity and distance from slot (One slot type hood with 1 in. slot)

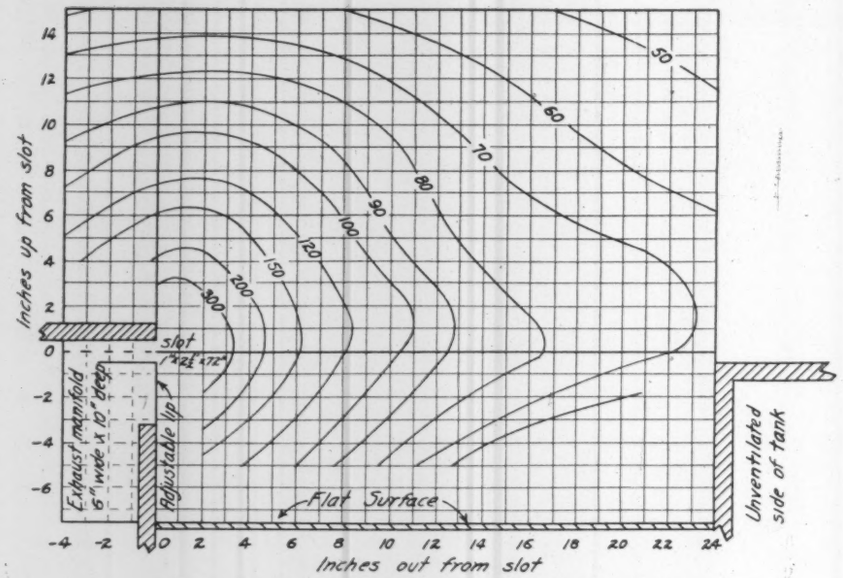

Fig. 2—Velocity contours for plating tank exhaust slot

= velocity at distance X from slot, and X = distance from slot at which V is determined.

However, in the case of a plating tank 6 ft long and 2 ft wide with a flat surface 7 in. below the center line plane of the slot the contour pattern will be distorted; that is, due to the restricted areas from which no air can be exhausted, the velocity contours in the plane normal to the slot at its midpoint will be expected to be extended or become separated adjacent to the zones of restriction. If this assumption is correct, it follows that V will vary inversely as X to some exponent less than one and that this exponent will not be constant unless the influence exerted by the restriction is uniform from slot outward X distance which is not the case with a plating tank. That is, if we substitute tank width W for distance from slot X in the equation given above, it becomes $Q = KLVW^a$ in which the value of "a" is less than one and is not constant for different tanks.

Results with One Hood

Since Dalla Valle's [Reference No. 5] general equation —

$$\frac{V}{V_o - V} = KX^a$$

gives the relation between the center line velocity V at various distances X from the exhaust opening and the velocity V_o through the opening, the values of the function

$$\frac{V}{V_o - V}$$

were plotted on double logarithmic paper against the corresponding values of X so that the value of the exponent "a" might be determined.

In Fig. 1 is shown the relation of the center line values for the function

$$\frac{V}{V_o - V}$$

to the corresponding values of X in inches. The resulting curve bears out the previous analysis. Since this relationship is not a straight line, it

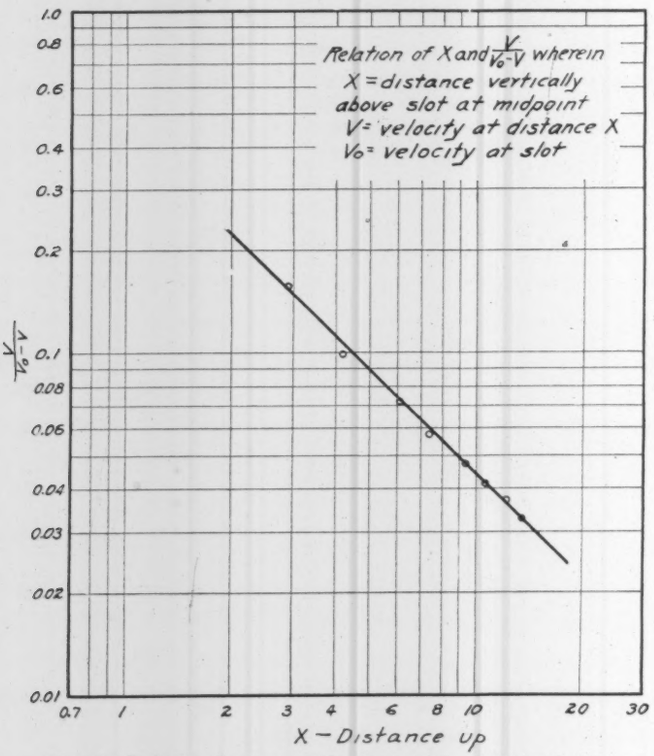
Relation of X and V/(Vo-V) wherein
X = distance vertically above slot at midpoint
V = velocity at distance X
Vo = velocity at slot
V/(Vo-V)
X - Distance up

Fig. 3—Relation between velocity in free zone and distance from slot (One slot type hood with 1 in. slot)

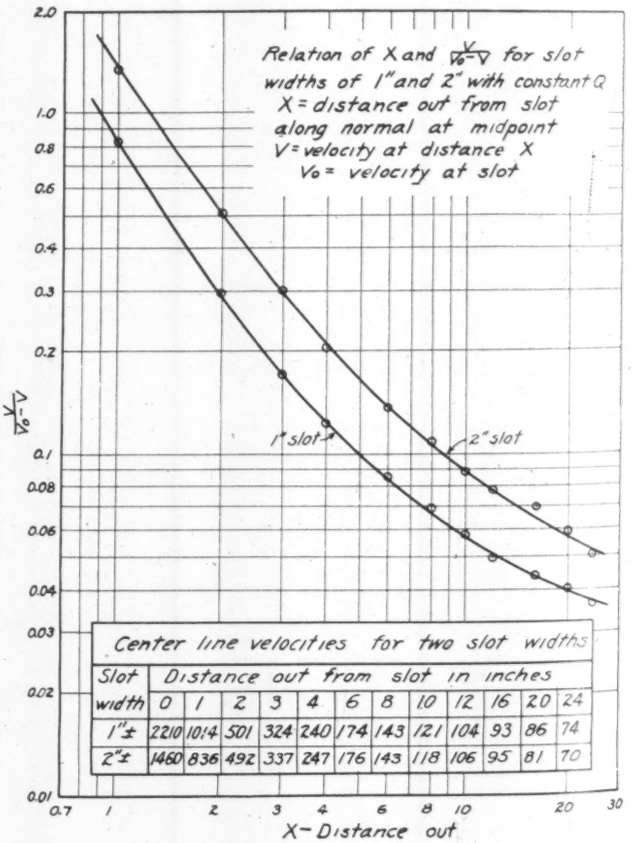
Relation of X and V/(Vo-V) for slot widths of 1" and 2" with constant Q
X = distance out from slot along normal at midpoint
V = velocity at distance X
Vo = velocity at slot
1" slot
2" slot
V/(Vo-V)
X - Distance out
Center line velocities for two slot widths
Slot width | Distance out from slot in inches
0 1 2 3 4 6 8 10 12 16 20 24
1"± 2210 1014 501 324 240 174 143 121 104 93 86 74
2"± 1460 836 492 337 247 176 143 118 106 95 81 70

Fig. 4—Effect of slot width upon center line velocity

is impossible to define it by means of any simple equation. However, from the point of view of controlling the gas and/or mist hazard in plating operations, the criterion seems to be a minimum air velocity toward the exhaust slot at the remote edge of the tank. Obviously, if the air velocity at the remote edge of the tank is adequate to carry the gas or mist into the exhaust system, at any point nearer the slot, the velocity will be more than adequate. Therefore, it may be desirable to estimate the value of the exponent "a" in the zone of minimum control velocity—that is, near the side of the tank opposite the hood. From the slope of the broken line at the lower end of the curve, this value is found to be about 0.50. Consequently the equation $Q = KLVW^{0.50}$ defines the relationship between the quantity of air to be exhausted and the center line velocity which will exist at the remote edge of the tank. While the minimum controlling velocity at the center line point most remote from the exhaust hood has not yet been determined, there is reason to feel that this value will be in the neighborhood of 75 fpm. If this value is correct, the equation for the calculation of the amount of ventilation necessary for tanks of about 12 to 30 in. in width and supplied with only one exhaust hood is $Q = 175\ LW^{0.50}$.

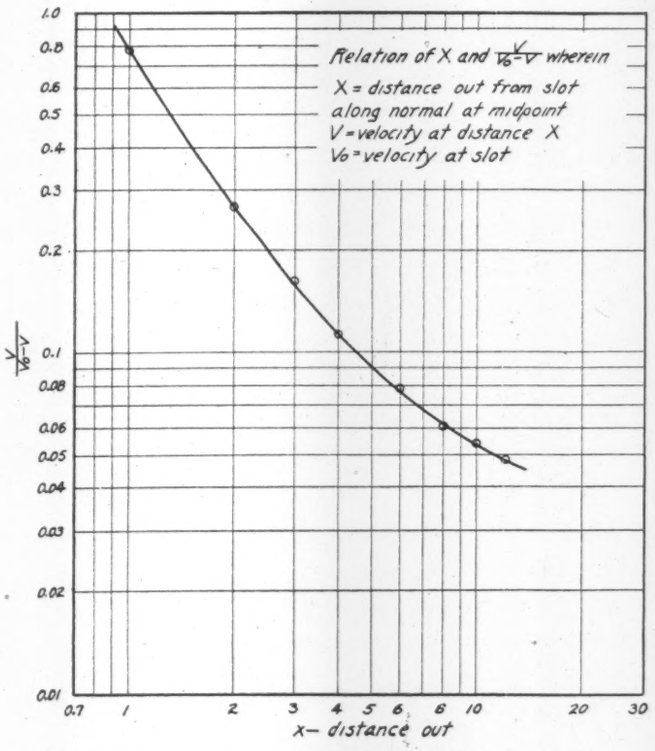

Fig. 5—Relation between center line velocity and distance from slot (Two slot type hoods with 1 in. slots)

It must be recognized, of course, that this equation may not be accurate for tanks of any width except 24 in. since the curve of Fig. 1 might not be the same for tanks of other widths inasmuch as any change in the location of the wall opposite the exhaust slot alters the velocity contours and the resultant curve to some extent. However, since tanks of greater width are usually provided with two exhaust hoods and for narrower tanks the error is on the safe side, it is felt that this equation will apply generally for tanks with one exhaust hood.

Since the equation $Q = 120LW$ has been used extensively and with good results as regards control of the contaminant, it may be well to see how the two equations compare. By the above equation for a tank 6 ft long by 2 ft wide $Q = 1440$ cfm. From the equation $Q = 175\ LW^{0.50}$, for a similar tank, $Q = 1488$ cfm. This same agreement does not exist, however, for tanks of all widths. For tanks less than 24 in. wide, the ventilation rate per unit width required by the new equation increases and for tanks over 24 in. it decreases—whereas with the equation now in use, it remains constant.

In Fig. 2 the velocity contours in a plane normal to the exhaust slot at its midpoint are shown. These contours demonstrate the distortion produced by the tank wall opposite the exhaust hood. This distortion improves the ventilation characteristics, since for a given Q the remote velocity contours extend beyond their normal path toward the critical velocity zone. In other words, if it were not for the wall on the unventilated side of the tank the 70 fpm contour would cross the center line about 14 in. from the slot instead of 23 in. from the slot and at 24 in. out or at the critical point the velocity would be only about 41 fpm.

As indicated in the introductory analysis, it is the tank wall on the unventilated side and to some extent the flat surface 7 in. below the slot which cause the exponent of X or W to be less than one and the relationship between

$$\frac{V}{V_o - V}$$

and X to be a curve rather than a straight line. This may be substantiated if the velocities as taken from the contours vertically above the slot in Fig. 2 are used in the function

$$\frac{V}{V_o - V}$$

and this function plotted against the corresponding distances above the slot. As shown in Fig. 3 a straight line having a negative slope of one will result.

Table 2—Comparison of Minimum Velocities (Fpm) Over Tank with One Slot and with Two Slots
(Quantity of Air Exhausted Same in Both Cases)

Number of Hoods	Distance from Slot, In.	Distance Above Center Line, In.			
		4	8	12	18
One	24	65	60	59	56
Two	12	65	70	70	61

Effect of Slot Width

While it has been shown by Dalla Valle [Reference No. 5] that slot or face velocity *per se* has little influence at even short distances from the hood, this fact is not generally recognized. The important factor is the quantity of air exhausted as is illustrated by the table in Fig. 4. This table gives the center line velocities at different distances from the midpoint of a 1 in. and 2 in. slot with Q remaining the same in both cases. These results indicate that the influence of the higher velocity is lost at a distance of 2 in. from the face of the slot. Obviously this distance will increase slightly as the difference between the slot widths increases. The apparent discrepancy between the slot velocities, as pointed out previously, is due to the fact that the adjustment of the slot widths is not very accurate so that they may vary substantially from the values given. However, this is of little consequence since the slot velocities were measured in all cases.

The curves of Fig. 4 are similar and essentially parallel as is to be expected.

Ventilation with Two Hoods

From the equation $Q = KVLW^{0.50}$ it might seem like a logical conclusion that since the ventilation rate required per unit tank width decreases with increasing width, less ventilation would be required for a tank with one hood than with two hoods. However, in the case of a tank provided with two hoods the resultant center line velocity at the center of the tank is the sum of the vectorial components of the velocities produced by both hoods, as was pointed out by Battista, Hatch and Greenburg [Reference No. 3]. While this value can be estimated with a fair degree of accuracy, it was felt desirable to make actual determinations of the center line velocity across a tank with two hoods so that these values, particularly the minimum control velocity, may be compared with that for a tank with one hood.

Results of Study with Two Hoods

The same tank was provided with exhaust ventilation through 1 in. slots along the two long sides and center line velocities were measured with a total ventilation rate of 1320 cfm (660 cfm through each hood). The relation between the function

$$\frac{V}{V_o - V}$$

and X is shown in Fig. 5. Since the ventilation characteristics are similar for both halves of the tank, values for only one-half of the tank are plotted.

There seemed to be so little difference in the minimum center line velocities obtained in the center of the tank with two exhaust hoods and at the remote edge with one hood, that the critical velocities vertically above the center line were determined at several stations for both conditions of ventilation. These results are shown in Table 2. While these results favor the two hoods only slightly, of far more importance is the practical significance: that is, that with one hood any movement by workers or objects near the unventilated side of the tank may cause sufficient conflicting air currents to permit the escape of some of the contaminated air, whereas with two hoods the velocity in the zone of probable movement is of such magnitude that no disturbance in the ventilation characteristics is likely.

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5) Dalla Valle, J. M.: Principles of Exhaust Hood Design, U. S. Public Health Service, Washington, 1939.

"OPEN FOR DISCUSSION"

I WANT TO COMPLIMENT

THE EDITOR—

I want to compliment C. M. Ashley on his article in the March HPAC entitled Air Change—Once a Major Criterion.

I have secured a number of copies of this paper and have sent them to a few of my friends with whom I have had this same argument over and over again for a number of years. Mr. Ashley has explained this subject so much better than I ever could; it has always been my opinion that the air change method was very obsolete.—W. J. DAVIS, Brandeis Machinery and Supply Co.

THE HIGH LIMIT OF AIR CHANGE

THE EDITOR—

In connection with the comments on Mr. Ashley's article on air change which have appeared in recent issues, perhaps it should be noted that it is not practical to handle unlimited volumes of air without draft, regardless of how well the grilles are selected or what type of grille is used.

Obviously, the volume of air required to be removed may be large enough to cause a general and uniform air movement exceeding that permitted by comfort standards. The maximum permissible number of air changes varies considerably with the occupancy or use of the space, its shape, location of supply and exhaust grilles, and velocity at which they operate, but a high limit for any type of room is approximately 30 air changes per hour.—D. J. STEWART, manager, electrical div., Barber-Colman Co.

Heating-Piping-Air Conditioning

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Journal Section

CONTENTS

JULY, 1941

The Interaction Constant for Moist Air, by John A. Goff and A. C. Bates 442

Interpretation of Summer Weather Temperature Data for Design, by J. Donald Kroeker and Harold V. Soballe 450

Local Cooling of Workers in Hot Industry, by F. C. Houghten, M. B. Ferderber and Carl Gutberlet 462

Local Chapter Reports 466

Candidates for Membership 472

Candidates Elected 472

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The Interaction Constant for Moist Air

By John A. Goff* and A. C. Bates**
Philadelphia, Pa.

Preliminary report of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in cooperation with the Towne Scientific School, University of Pennsylvania

SUMMARY — Preliminary measurements on the interaction constant for moist air are reported. This constant evaluates the effect of intermolecular forces between the air and water vapor. Its magnitude at 68 F indicates a deviation from Dalton's Law of 0.52 per cent which, though small, is nevertheless significant

Introduction

IN Chapter 1 of THE GUIDE 1941 it was explained that an accurate and thermodynamically consistent formulation of the properties of moist air awaits the experimental measurement of a certain temperature function A_{aw}, called the interaction constant, expressing the effect of forces between air and water molecules. It is realized, of course, that there is no such thing as an air molecule, since air is itself a mixture of several gases; but the justification for regarding this mixture as a single constituent was also explained in THE GUIDE†.

The present paper is a preliminary report on the experimental investigation being conducted at the Towne Scientific School, University of Pennsylvania, under a cooperative agreement with the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS through its Research Technical Advisory Committee on Psychrometry††. The immediate objective of the investigation is the measurement of the interaction constant A_{aw} over the range of temperature of interest to air conditioning. To obtain satisfactory results by the method employed it was necessary to design a rather elaborate apparatus with which measurements could be made to an accuracy of at least 0.1 per cent. Satisfactory results have been obtained for only one temperature, namely 15C, but it is felt that a preliminary report outlining the theory of the experiment, describing the apparatus, and discussing present results will be of interest.

Theory

The method employed may be described briefly as the *saturation isotherm* method. A steady stream of dry, carbon dioxide-free air is passed through a saturator at high pressure P_1, then through a drying train in which m_1 pounds of water is removed, then through a second saturator at low pressure P_2 and finally through a second drying train in which m_2 pounds of water is removed. Both saturators are immersed in the same thermostat whose temperature T is closely regulated and accurately measured. Assuming (1) complete saturation in the saturators, (2) complete drying in the drying trains, (3) negligible leakage, (4) correctness of theory, and (5) accuracy of all measurements, the five data P_1, P_2, m_1, m_2, T suffice for the determination of the interaction constant A_{aw} without requiring to weigh the quantity of dry air passed through the apparatus.

In order to provide a check on the above assumptions, it was arranged to weigh the dry air. Preliminary saturators in a separate thermostat were provided so that saturation in the main thermostat could be approached either by condensation or by evaporation. Suitable reducing valves were provided to permit changing the pressures P_1 and P_2 in order to make sure that the results are independent of pressure and dependent only on temperature as required by theory. Orifices of different sizes were provided for the discharge line in order to be able to vary the rate of flow through the apparatus.

The theory of the experiment is built upon the prediction of statistical mechanics expressed in Equation (9) of THE GUIDE, namely,

$$Pv = RT - [A_{aa}x^2 + 2A_{aw}x\,(1-x) + A_{ww}\,(1-x)^2]P; \quad \ldots (1)$$

where

P = observed pressure
v = specific volume (volume per mol)
A_{aa} = second virial coefficient for dry air, a known temperature function
A_{ww} = second virial coefficient for water vapor, a known temperature function
A_{aw} = the interaction constant, the temperature function to be measured
x = mol fraction of dry air in the mixture

Applying a well-known identical relation of thermodynamics to (1) and observing a further prediction of statistical mechanics in adjusting the constant of integration, the following expressions for the chemical potentials are easily derived:

$$\mu_a = RT \log_e (xP) + [(A_{aa} - 2A_{aw} + A_{ww})\,(1-x)^2 - A_{aa}]\,P + g_a\,(T), \quad \ldots (2a)$$

$$\mu_w = RT \log_e [(1-x)\,P] + [(A_{aa} - 2A_{aw} + A_{ww})\,x^2 - A_{ww}]\,P + g_w\,(T) \quad \ldots (2b)$$

where $g_a\,(T)$ and $g_w\,(T)$ are temperature functions which will not need to be evaluated explicitly.

As explained in THE GUIDE (p. 18), the conditions for saturation are equality, between the two coexisting phases, of (*a*) pressure (*b*) temperature and (*c*) each component chemical potential. Therefore, in order to set up the equation for the

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†In this and subsequent references to THE GUIDE, Chapter 1 of THE HEATING, VENTILATING, AIR CONDITIONING GUIDE, 1941 is intended.

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saturation isotherms of moist air, suitable expressions for the chemical potentials of dry air and water in the liquid phase must first be written down.

Now the saturated liquid phase will be almost pure water, but containing a very small concentration of dissolved air. This dissolved air will not have the same composition as that of dry air, but will show considerable oxygen enrichment. However, this effect is entirely negligible and it is legitimate to regard the liquid phase as an *ideal solution* of dry air in water. Furthermore, in the range of pressures investigated the liquid phase may also be regarded as incompressible; whence the appropriate expressions for the chemical potentials are[1]

$$\mu'_a = RT \log_e x' + \bar{V}'_a P + g'_a(T), \quad \ldots\ldots (3a)$$

$$\mu'_w = RT \log_e (1 - x') + v'_w P + g'_w(T); \quad \ldots\ldots (3b)$$

where

x' = mol fraction of dissolved air,

$\bar{V}'_a$ = partial molal volume of dissolved air,

v'_w = specific volume of pure liquid water,

and $g'_a(T)$, $g'_w(T)$ are temperature functions which will not need to be evaluated explicitly. The prime refers to the liquid phase.

Equating (2b) and (3b) as one condition for saturation, and then noting that x and x' must vanish simultaneously when the pressure P is reduced to the saturation pressure p_s of pure water for the given temperature, the result is

$$RT \log_e \left[\frac{(1-x)P}{p_s}\right] = RT \log_e (1 - x') + (v'_w + A_{ww})(P - p_s) - [(A_{aa} - 2A_{aw} + A_{ww})x^2] P. \quad \ldots (4a)$$

Next, equating (2a) and (3a), as a second conditon for saturation,

$$RT \log_e \left[\frac{x'}{xP}\right] = -\bar{V}'_a P + [(A_{aa} - 2A_{aw} + A_{ww})(1-x)^2] P + g_a(T) - g'_a(T). \quad \ldots\ldots (4b)$$

The mol fraction x' is to be eliminated between these two equations in order to obtain the relation between mol fraction x and pressure P for a given temperature.

Fortunately, the required elimination can readily be effected by the use of approximations which can be validated later. In the first place, the term $RT \log_e (1-x')$ may be replaced by $-RT\,x'$, since x' is so very small; and, this being so, the actual dependence of the righthand member of (4b) on pressure and mol fraction x may be ignored and this equation reduced to the form of Henry's Law for dilute solutions, namely,

$$\frac{x'}{xP} = k(T). \quad \ldots\ldots (5)$$

Finally, it may be anticipated that the righthand member of (4a) will be small enough to justify the use of the approximation, $1-x = p_s/P$, for the purpose of evaluating (5) and the rightmost term of (4a). The final result after dividing through by RT, is

$$\log_e \left[\frac{(1-x)P}{p_s}\right] = A\left(\frac{P}{p_s} - 1\right) - B(1-\lambda)\left[\frac{P}{p_s} - 2 + \frac{p_s}{P}\right]; \quad \ldots\ldots (6)$$

where

$$A = k p_s + \frac{v'_w p_s}{RT} + \frac{A_{ww} p_s}{RT} \quad \ldots (7a)$$

and

$$B = \frac{A_{aa} p_s}{RT} + \frac{A_{ww} p_s}{RT} \quad \ldots\ldots (7b)$$

are dimensionless temperature[2] functions which can be calculated from data available in the literature; and where

$$\lambda = \frac{2A_{aw}}{A_{aa} + A_{ww}} \quad \ldots\ldots (8)$$

is introduced as a dimensionless interaction constant which, according to theory, should depend on temperature only. In what follows, attention will be directed to the determination of λ rather than of A_{aw} itself.

It will be noticed that, if λ were known for a given temperature, the mol fraction x could be calculated from (6) for various values of pressure P. From the values thus obtained, values of mol *ratio*, $n_w/n_a = (1-x)/x$, could be calculated and multiplied by $(P/p_s - 1)$. In this manner a family of curves of which three are shown in Fig. 1 could be determined for any given temperature. Suppose, then, that the *same* weight of dry air n_a (mols) were saturated first at pressure P_1 and then at lower pressure P_2 but the *same* temperature. The difference between values of the ordinate in Fig. 1 between the two pressures would be

$$\log_e \left[\frac{m_1 (P_1 - p_s)}{m_2 (P_2 - p_s)}\right],$$

where the weight of air cancels out and m_1 and m_2 are the weights of water (in pounds or grams) removed after each saturator. But this difference in ordinate can be measured directly: hence the determination of λ for the given temperature reduces to picking that curve of Fig. 1 showing the measured difference in ordinate between the measured pressures. Actually, this procedure was formulated mathematically so that values of λ could be calculated directly without resort to graphical representation. This was an important step in expediting routine calculations, but need not be given here.

It is desired to emphasize that the above method of determining λ does not require to know the weight of dry air passing through the apparatus. However, the weight of dry

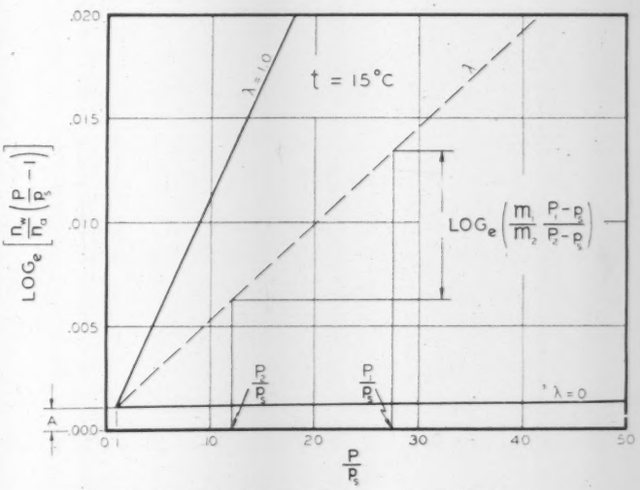

Fig. 1—Graphical determination of *lambda*

[1]Numbers refer to references listed in Bibliography.

air was actually measured so that measured values of mol fraction x were available for substitution in (6) as a check. At first this check failed by an amount which could be attributed to an error of about 0.08 C in the measurement of temperature. The thermometer used had been calibrated by the Bureau of Standards in 1924; but, since corrections were given at 20-degree intervals only, it was necessary to interpolate for a correction at 15 C. While the thermometer was being recalibrated at the Bureau of Standards, the apparatus was carefully examined for leaks only to find them entirely negligible; also minor changes were made to improve pressure regulation. When the thermometer was returned after recalibration, the actual correction at 15 C was found to be almost exactly what was required to provide a check with the measured weight of dry air through (6). This was most gratifying.

Experimental Method and Apparatus

The choice of experimental method, for which the theory of the preceding section was developed, was based on a study of the difficulties likely to be encountered in alternative methods—such as measuring simultaneous values of pressure, specific volume, temperature and composition and calculating A_{aw} directly from (1)—and the consideration that a direct determination of the weight of water per pound of dry air *at saturation* would be of special interest to the air conditioning engineer. Before attempting to design the apparatus, however, it was necessary to decide what range of temperature was to be covered, at what pressures it was feasible to operate, what precision in each measured quantity would be required in order to obtain significant values of λ.

The lower limit of temperature was easy to decide upon as 0 C, since below this the condensed phase would be ice which would dictate a radically different saturator design. The upper limit was taken to be 30 C. Preliminary consideration of the curvature of the saturation isotherms at low pressures according to (6) suggested $P_2 = 10\ p_s$ as lower limit for pressure. The upper limit was taken to be $P_1 = 200\ p_s$. Keyes[2] had estimated that at atmospheric pressure saturated air at 30 C and at 40 C would contain about 1 per cent more water vapor than the ordinary application of Dalton's Law would predict. This made it appear necessary not to allow any measured quantity to be uncertain by more than 0.1 per cent. The severity of this precision requirement will be the better appreciated when it is realized that at 0 C, 10 p_s is only 1.85 in Hg and 0.1 per cent of this is only 0.002 in Hg. In order to attain accuracy of this sort, automatic control and regulation of temperature and pressure over comparatively long test periods and painstaking attention to all details were absolutely necessary. The design calculations, the problems of matching and compromising equipment characteristics, and preliminary tests on component parts of the apparatus cannot be discussed in this report for lack of space. Suffice it to say that the demands for inventiveness, physical intuition, and mechanical skill were considerable and explain the apparent slowness in obtaining even preliminary results.

Fig. 2 is a flow diagram of the

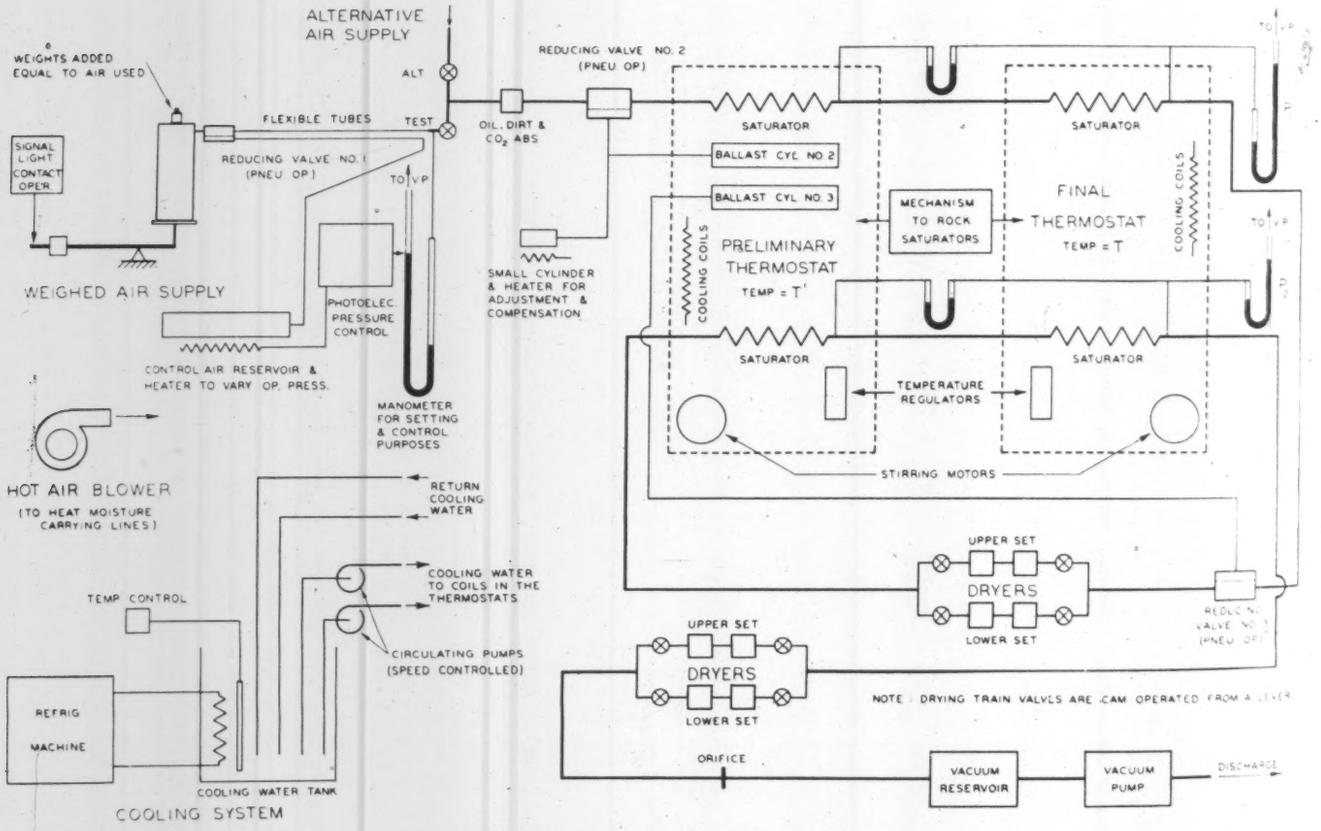

Fig. 2—Schematic diagram of the test equipment

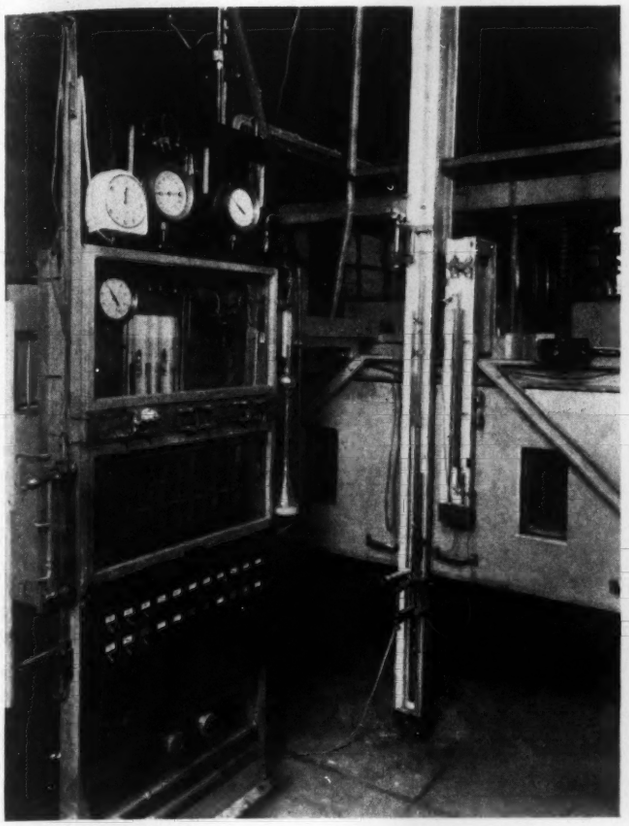

Fig. 3—General view

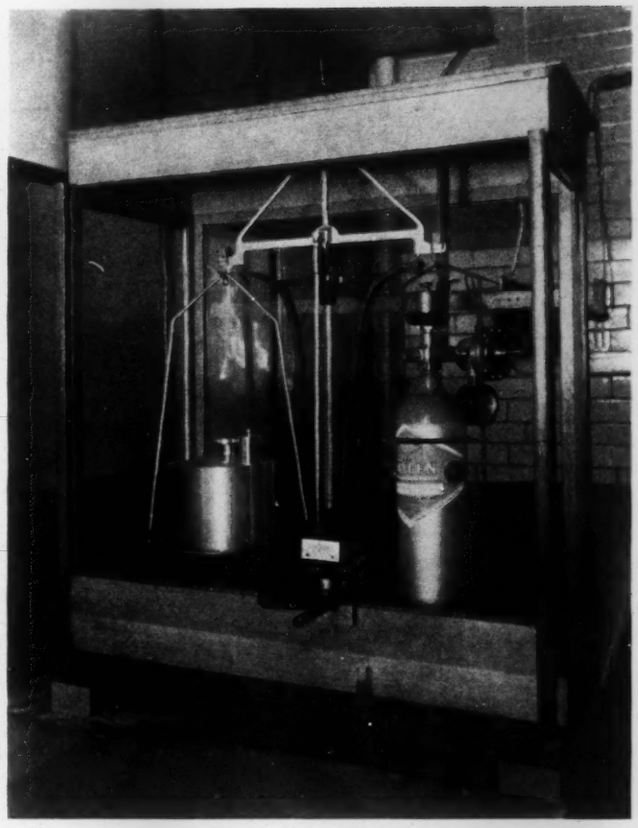

Fig. 4—Weighed air supply

apparatus in its present form and represents an attempt at completeness without loss of simplicity. The photographs of Figs. 3 to 6 give an idea of appearance, arrangement, proportions and some details.

During actual test periods the supply of air comes from a small cylinder initially containing from 150 to 220 grams of air depending on how many times it has been filled from a standard 200 cubic foot cylinder. When it is no longer possible to fill the small cylinder to a pressure of 1500 lb per square inch, a fresh large cylinder is used, the air remaining in the old one being used for pre-test and in-between runs, for charging the pneumatic controls, and for general utility purposes.

The small test cylinder depends from one arm of a sensitive balance. In order to limit the pressure in the take-off tube and keep it reasonably constant, a small reducing valve was mounted directly on the cylinder. This valve was modified for pneumatic control from the main control panel, which obviated the necessity of disturbing the balance except to add weights, but required an additional tube paralleling the take-off tube. Weights were added directly over the center of gravity of the cylinder so as not to alter the position of the cylinder relative to the tubes. In order to eliminate correction for buoyancy, a dummy weight made of aluminum and having about the same density as that of the cylinder and its contents was placed on the other scale pan. Although some minor improvement can yet be made in the air weighing apparatus, nevertheless, in its present form it has been proven not to introduce an error of as much as 0.1 per cent of the total weight of air withdrawn during the test period. As suggested above, weighing was done by the method of substitution, balance being indicated by electrical contact.

Owing to the fact that the pressure in the test cylinder drops continually as air is withdrawn during test, the first reducing valve requires some compensation. To provide this a mercury manometer was connected to the delivery line. On the manometer was mounted a photoelectric cell which operated through a suitable relay to heat the air on the control side of the valve thus raising the control pressure and compensating for the drop in supply pressure.

The air delivered by the first reducing valve passed through a CO_2 remover, then through a second reducing valve, also pneumatically controlled, then through the high pressure saturator in the preliminary thermostat to the high pressure saturator in the main thermostat. The second reducing valve was for the purpose of closer pressure regulation.

The saturators were patterned after those used by Gordon[3] and Washburn[4] but were constructed from ½ in. O.D., 0.035 in. wall tubing, with solder type fittings instead of glass, and modified by the addition of a filling manifold and gauge glasses. The use of copper was dictated by pressure and heat transfer requirements. After Test No. 3 the rocking speed of the saturators was reduced to about once in forty seconds it having been noted that, at the higher speed of once in fifteen seconds used in previous tests, slight pressure pulsations occurred due to the formation of a film causing momentary interruption of the flow. Connections to the saturators were made of copper tubing designed to be leak proof and self draining, with adequate fatigue life. All lines to and from the saturator

were heated by air from an ordinary hair dryer to prevent condensation. This warm air was carried over to the glass-enclosed control panel where various valves carrying moist air were located.

In all tests after Test No. 3, anhydrone was used as drying agent. The usual procedure of providing alternative drying sets of approximately matched flow resistances, switching from one set to the other at start and finish of each test, was followed. The switch-over valves were cam operated from a single cam shaft, care being taken not to turn the shaft too quickly for fear of carrying some of the finely divided anhydrone out of the drying towers. Other precautionary measures were taken in the handling of the dryers which need not be described in detail here.

Returning to the flow diagram, the air stream from the high pressure saturator was sent through a third reducing valve before entering the first drying train in order to obviate the necessity of designing the towers for the high pressure. It then passed through the low-pressure saturator in the preliminary thermostat, then through the low-pressure saturator in the main thermostat, then through the second drying train, and finally discharged into an evacuated carboy through a suitable orifice to regulate the rate of flow.

For measuring the pressures at exit from the two main saturators, mercury manometers with one leg connected to a high-vacuum pump were used, the absolute pressure being measured by a McLeod gauge. This arrangement made the readings of the manometers independent of changes in barometric pressure. Excessive trouble was experienced in keeping the mercury menisci clean until one-piece glass construction was adopted. It is felt that ultimate precision in the measurement of pressure has not yet been reached and some changes will be made before additional tests are started.

The temperature regulators were constructed along conventional lines. They operated through thyratron tubes which start and stop suitable pumps supplying refrigerated water (with a suitable antifreeze added to prevent freezing and improve heat transfer) to the cooling coils in the thermostats. The evaporation coil of the small refrigeration machine was placed in a separate insulated tank partially filled with the cooling medium. This arrangement not only made for good control, but allowed for the possibility of inserting an electric heater in experiments at higher temperatures. The temperature regulation at 15 C was about ±0.001 C as determined by a Beckmann thermometer. Closer regulation, though obtainable, was not considered necessary.

The pneumatic operation of all reducing valves makes them insensitive to changes in barometric pressure, permits their use at pressures above or below atmospheric, and makes delicate adjustment and automatic control possible. The extent of the electrical auxiliaries may be indicated by mentioning that the apparatus includes twenty-two switches, two thyratron tubes, and one photoelectric cell plus a relay tube.

Test Procedure

A detailed explanation of the test procedure will be omitted from this report but will be found in the

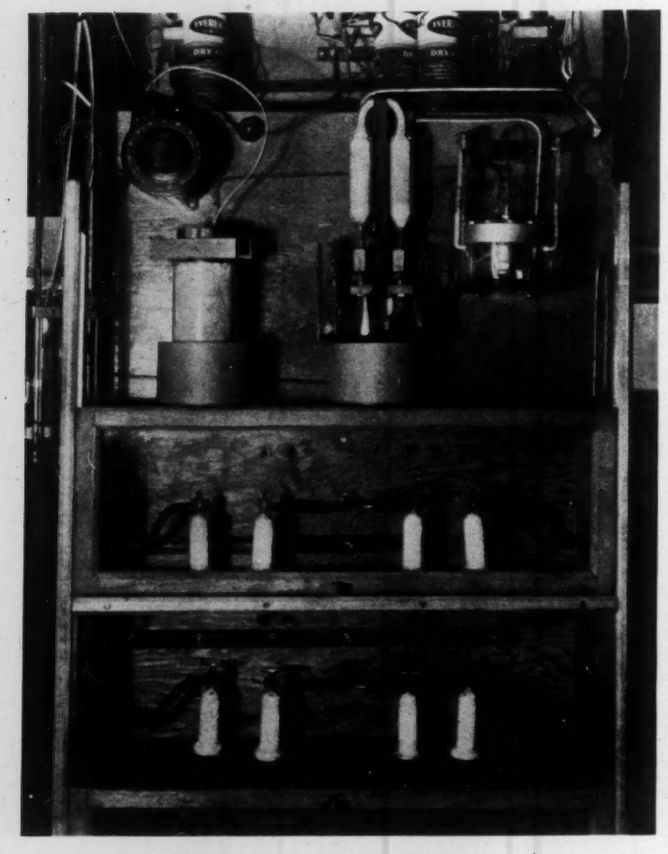

Fig. 5—Dryers, back of panel

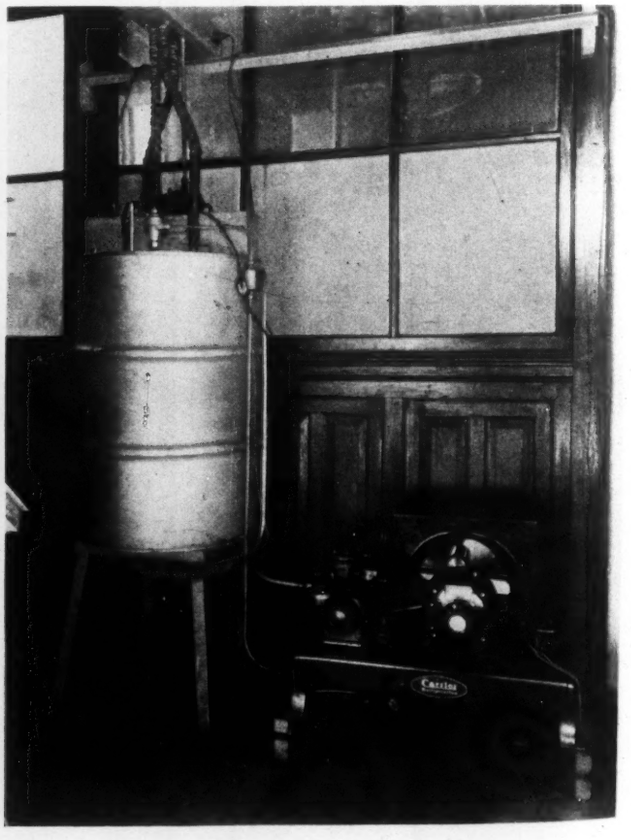

Fig. 6—Cooling system

thesis††† submitted by one of the authors to the Faculty of the Graduate School of Arts and Sciences, University of Pennsylvania, in partial fulfillment of the requirements for the doctor's degree in engineering. A copy is on file in the Towne Scientific School Library.

Test Data

Test No. 1 is to be rejected because of incomplete drying. Test No. 8 is also rejected because of unfortunate failure to record the initial weight of one of the drying towers. Table 1 lists the data obtained in all other tests so far performed. The temperature was the same in all tests, namely 14.91 C.

Table 1—Test Data

1	Test Number	2	3	4	5	6	7	9
2	Temperature, main thermostat (deg C)	14.91	14.91	14.91	14.91	14.91	14.91	14.91
3	Temperature, preliminary thermostat (deg C)	14.9	20	14.9	14.9	14.9	14.9	14.9
4	High Pressure (in. Hg)	50.522	50.573	50.630	50.413	50.490	25.395	25.272
5	Low Pressure (in. Hg)	25.266	25.272	25.265	25.261	25.356	12.560	12.594
6	Weights added to scales (grams)	150	160	115	170	170	85	170
7	Duration of Test (hours)	3.88	4.11	2.97	4.39	4.37	4.51	9.33
8	Nominal Rate of Flow (grams per hr)	40	40	40	40	40	20	20
9	Weight water, first dryer (grams)	0.9542	1.0099	0.7247	1.0744	1.0732	1.0733	2.1581
10	Weight water, second dryer (grams)	1.9001	2.0254	1.4581	2.1567	2.1479	2.2115	4.4101
11	Corrections: (a) Change in control press.	+0.002	+0.005	+0.004	+0.006	+0.004	+0.001	+0.004
	(b) Change in delivery press.	+0.017	+0.006	0.000	−0.006	0.000	+0.026	0.000
	(c) Dirt, H_2O, CO_2	−0.090	−0.096	−0.069	−0.102	−0.102	−0.051	−0.102
	(d) Leakage	———	———	−0.018	−0.026	−0.026	−0.027	−0.056
	(e) Net	−0.071	−0.085	−0.083	−0.128	−0.124	−0.051	−0.154
12	Corrected weight of dry air	149.93	159.91	114.92	169.87	169.88	84.95	169.86

Changes in control pressure and delivery pressure alter slightly the quantity of air stored in the first reducing valve so that the loss of weight of the test cylinder assembly is not exactly the weight of air reaching the main apparatus. Special preliminary tests were performed to determine the necessary corrections (items 11a and 11b). The correction for H_2O in the supply air (item 11c) was computed by assuming that the air in the large supply cylinder is continuously saturated during the process of filling the small test cylinder. The correction for dirt and CO_2 was determined by passing a weighed sample of air through the CO_2 remover and noting the gain in weight (item 11c). Leakage corrections were inferred from pressure readings over extended periods under shut-off conditions (item 11d). For Tests 2 and 3, no leakage corrections were determined.

†††Doctoral Dissertation, University of Pennsylvania, 1941. "Measurement of the Interaction Constant for Moist Air by the Saturation Isotherm Method," by Arthur Crocker Bates.

Reduction of Test Data

In reducing the data of Table 1, the thermal data given in Table 2 as obtained from the literature were used.

Table 2—Thermal Data from the Literature

t C	p_s (in. Hg)	kp_s x10²	$v'_w p_s/RT$ x10³	$A_{ww}p_s/RT$ x10³	$A_{aa}p_s/RT$ x10⁶	A x10³	B x10³
0	0.18110	1.42	0.485	0.5609	3.598	0.5656	0.5645
5	0.25853	1.83	0.676	0.7081	4.611	0.7147	0.7127
10	0.36385	2.25	0.94	0.8850	5.669	0.8942	0.8907
15	0.50529	2.83	1.28	1.0956	7.161	1.1081	1.1028
20	0.69290	3.55	1.72	1.3449	8.676	1.3618	1.3526
25	0.93884	4.53	2.29	1.6370	10.301	1.6594	1.6473
30	1.25774	5.51	3.06	1.9780	11.978	2.0080	1.9900
14.91	0.50236	2.82	1.28	1.0915	7.136	1.1042	1.0986

Values of the saturation pressure p_s are Bureau of Standards[5] values converted to inches of mercury at 68 F, $g = 980.196$ (Philadelphia). Values of kp_s were computed from solubility data in International Critical Tables. Values of $v'_w p_s/RT$ were computed from steam table data.[5] Values of $A_{ww}p_s/RT$ were calculated from the formula of Keyes.[6] Values of $A_{aa}p_s/RT$ were calculated from the data of Holborn and Otto[7] and from the formula of Beattie and Bridgeman,[8] as modified by Keyes.[2] The quantities A and B were defined in terms of the preceding quantities by Equation (7).

Table 3 shows the final reduction of the data of Table 2. Values of the interaction constant λ (item 6) are computed without using the measured weight of dry air in the manner previously explained. With λ determined in this manner the mol fraction of water vapor $(1-x)$ is computed from (6) for either high or low pressure (item 2 or item 3); then using the corresponding measured weight of water (item 4 or item 5), the weight of dry air is calculated (item 7). Item 9 is the difference between item 8 and item 7 expressed in per cent.

Little significance is assigned Tests 2 and 3 except as they indicate that saturation was being closely approached by showing that whether the preliminary thermostat was set for 15 C or 20 C had little effect. The reasons for discounting these tests are certain irregularities in the performance of the apparatus and unsatisfactory drying. In Test 4, it was forgotten to start the rocking motor at the beginning of the test; therefore, it was left off during the entire test in order to see the effect upon the results.

Tests 5, 6, 7, 9 are considered the only ones that should be used in computing the most probable value of λ. Item 10 shows that these four tests are remarkably consistent showing an average failure to cross-check the weight of dry air of only −0.08 per cent. This might be due to a residual temperature error of 0.012 C which could be merely an interpolation error since the thermometer was graduated only to tenths. On the other hand, there might be a consistent error in the air weighing; and in future tests some improvements calculated to reduce

Table 3—Reduced Test Data

1	Test Number	2	3	4	5	6	7	9
2	High Pressure Ratio (P_1/p_s)	100.569	100.671	100.784	100.352	100.506	50.551	50.307
3	Low Pressure Ratio (P_2/p_s)	50.295	50.307	50.293	50.285	50.474	25.002	25.070
4	Weight water, first dryer (grams)	0.9542	1.0099	0.7247	1.0744	1.0732	1.0733	2.1581
5	Weight water, second dryer (grams)	1.9001	2.0254	1.4581	2.1567	2.1479	2.2115	4.4101
6	Interaction Constant	0.253	0.138	0.105	0.072	0.085	0.064	0.082
7	Calculated Weight Dry Air (grams)	148.33	159.14	114.74	169.99	169.82	85.09	170.08
8	Corrected Weight Dry Air (grams)	149.93	159.91	114.92	169.87	169.88	84.95	169.86
9	Per cent Deviation	+1.07	+0.48	−0.16	−0.07	+0.04	−0.16	−0.13
10	Average Deviation				−0.08%			
11	Drying Agent	Activated Alumina			Anhydrone			

this uncertainty will be made. Also some indicated improvements in the measurement of pressure will be made since the present failure to check might be explained by errors in pressure of a few thousandths of an inch of mercury.

The variation in the values of λ for the last four tests as given in Table 3 might at first blush appear disappointing. However, it must be remembered that extreme precision would be required to reduce this variation below about 5 per cent. After careful consideration of all recognizable sources of error in the individual tests and weighting the results accordingly, it appears safe to say that

at 14.91 C, $\lambda = 0.075 \pm 0.005$

Reduction of Pollitzer and Strebel Data

So far as the authors are aware, the only published data with which to compare the results of this investigation are those of Pollitzer and Strebel,[9] at 50 C and 70 C. These measurements are by no means of comparable precision with those presented here; but they yield fairly good values of λ because, at the much higher pressures employed, the precision requirements are much less severe. Thus the average departure from Dalton's Law for the range of pressure employed by Pollitzer and Strebel is about 30 per cent instead of 0.6 per cent for the range covered in the present experiments. Offsetting this advantage to an unknown extent, however, is the fact that the theory available at the present time is limited to the range of pressures in which Equation (1) is applicable.

Two methods were used in reducing the data of Pollitzer and Strebel. In the first, a value of λ was computed for each test using Equation (6). Assuming no systematic error, an attempt was made to select intervals of λ which would give a normal frequency distribution. This can be done for the 50 C tests by disregarding low pressure points which are obviously erratic; but no choice of intervals could be found to make a normal distribution of errors for the 70 C tests. The most probable values of λ determined in this way are judged to be: at 50 C, $\lambda = 0.057 \pm 6$ per cent; at 70 C, $\lambda = 0.087 \pm 16.5$ per cent.

The second method is to apply least squares to determine λ and a factor ϵ by which the nominal saturation pressure p_s might be multiplied to account for a small error in measurement of temperature and/or weight of air. Again omitting low pressure points, the results are:

at 50 C, $\lambda = 0.052$ and $\epsilon = 1.024$; at 70 C, $\lambda = 0.065$ and $\epsilon = 1.033$.

After careful consideration the following results were selected as representing the best reduction of the Pollitzer and Strebel data possible without employing methods involving computational labor entirely unjustified by any use to be made of the results

50 C: $\lambda = 0.056$
70 C: $\lambda = 0.07$

Fig. 7 shows the desired comparison with present results. No curve is drawn, but there is some indication that λ does not vary markedly with temperature and may be assumed to have a constant value of about 0.075 pending further experimentation.

Discussion of Results

If the object of the present investigation were merely to adjust λ so as to make Equation (6) fit the experimental data, very little discussion beyond that already given would be required. However, it is proposed to use the values of λ inferred from these experiments, and from those at different temperatures to follow, for the purpose of computing specific volume by means of Equation (1) and specific enthalpy from the following equation obtained by applying a well-known identical relation of thermodynamics to Equation (2):

$$h = xh^\circ_a + (1-x)h^\circ_w + [B_{aa}x^2 + 2B_{aw}x(1-x) + B_{ww}(1-x)^2]P \quad \ldots (9)$$

where

h°_a = zero-pressure specific enthalpy of dry air obtained from spectroscopic data (Table 3, THE GUIDE)

h°_w = zero-pressure specific enthalpy of water vapor obtained from spectroscopic data (Table 5, THE GUIDE)

and where, for example,

$$B_{aw} = -T^2 \frac{d(A_{aw}/T)}{dT} \quad \ldots (10)$$

Therefore it is important to adduce as much evidence as possible for believing that the value of λ determined from (6) is indeed the interaction constant to be used in (1) and (9).

In the first place, it is important to adduce evidence of the complete saturation assumed in (6) or, alternatively, evidence of independence of the final results on whether complete saturation is attained or not. For example, theory[10] indicates that, at constant temperature, the degree of saturation depends only on the weight of air flowing through the saturators per unit time. The weight flowing through each pressure stage being the same in any one test, the same degree of saturation should be reached in both saturators and the ratio m_1/m_2, on which the determination of λ depends, should be correct even if complete saturation were not attained. Of course, incomplete saturation would cause failure to cross-check with the measured weight of air through (6); however, the average deviation of —0.08 per cent is of the wrong sign to be explained by incomplete saturation, though this fact in itself does not completely exclude the possibility. The higher rate of flow in Tests 5 and 6 would be expected to result in lower values of $(1-x)P/p_s$ for these tests than for Tests 7 and 9. To assume this effect just sufficient to bring the points for the former into agreement with those for the latter at $P/p_s = 50$ would amount to assuming that doubling the rate of flow lowered the degree of saturation by 0.1 per cent. Making this assumption would reduce somewhat the probable error in λ, but at the same time would increase numerically the average deviation between measured and calculated weight of air. Additional tests with the preliminary thermostat at higher temperature, plus tests at a third rate of flow, might explain present discrepancies. Therefore the limits of uncertainty in the value of λ have been left as determined without indulging in assumptions.

Table 6—Thermodynamic Properties of Moist Air, 29.921 In. Hg

Temp Deg F	Saturation Humidity Ratio W_s Weight of Water per Lb of Dry Air		Volume Cu Ft per Lb Dry Air			Enthalpy Btu per Lb Dry Air			Specific Enthalpy of Liquid Water Btu per Lb h'_w	Saturation Pressure p_s		Temp Deg F
	Pounds	Grains	Dry Air v_a	v_{as} $(v_s - v_a)$	Saturated Mixture v_s	Dry Air h_a	h_{as} $(h_s - h_a)$	Saturated Mixture h_s		In. of Hg	Lb per Sq In.	
68	0.01468	102.8	13.29	0.31	13.60	16.31	16.00	32.31	36.0	0.68980	0.3388	68
68	*0.01477*	*103.40*	*13.299*	*0.314*	*13.613*	*16.324*	*16.080*	*32.413*	*36.04*	*0.69413*	*0.3409*	*68*

Conclusions

The following conclusions are considered reasonable: (1) the theory is adequate, at least in the low-pressure range, since all deviations can be ascribed to residual experimental error; (2) the two-stage saturation isotherm method gives exceedingly consistent results; (3) weighing the air is practically indispensable as a cross-check until satisfactory operation of the apparatus is established; (4) the possibility of measuring λ by the saturation isotherm method is established; (5) aside from the precision of all direct measurements, the two most important experimental items are elimination of leakage and extremely close pressure regulation; (6) now that the theory has been checked in the low pressure range, future tests may be made at higher pressures where it is easier to determine λ accurately even though it may then be impossible to weigh the air on account of the limited capacity of the present weighing apparatus.

Summary

Preliminary measurements of the interaction constant for moist air by the saturation isotherm method are reported. The value $\lambda = 0.075 \pm 0.005$ for 15 C is regarded as the best reduction of the data obtained. Comparison with certain data in the literature indicates that λ will be found not to vary markedly with temperature. This would correspond to a Dalton factor, at standard atmospheric pressure and 20 C, of 1.0052 which is much smaller than the value guessed in The Guide 1941 but still much larger than the Poynting factor and very much larger than the Raoult factor, two corrections usually applied to the vapor pressure. The indicated revisions of the data in Table 6, The Guide 1941, at 20 C are shown above.

Acknowledgment

Acknowledgment is made to Dr. F. R. Bichowsky, Chairman of the ASHVE Research Technical Advisory Committee on Psychrometry, who was instrumental in enlisting the support of the Society and whose advice on experimental matters has contributed greatly to the success of the investigation; also to Dr. W. H. Carrier for his cooperation in lending a small refrigeration machine used in connection with this investigation.

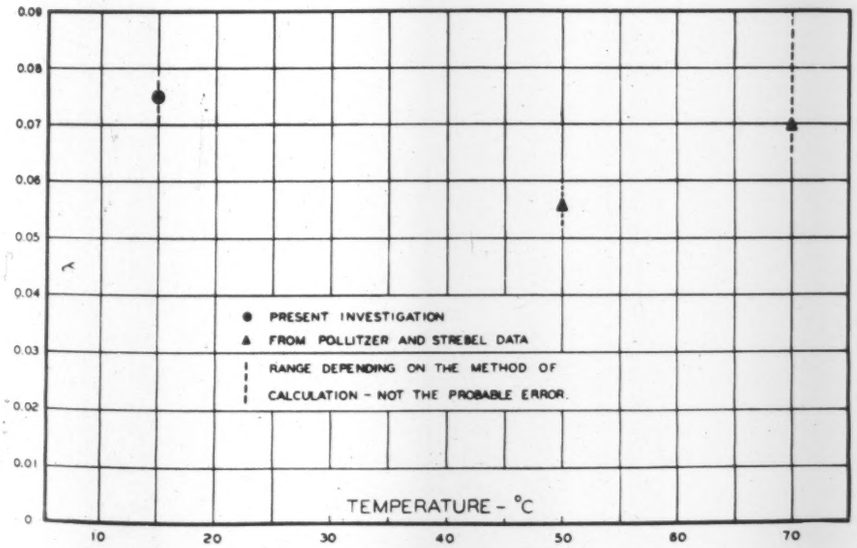

Fig. 7—Available values of *lambda*

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THE MILL ON THE HILL

In the Pittsburgh area more than 4,300,000 cubic yards of earth and rock were moved in creating the 300-acre man-made plateau where a modern sheet and tin-plate mill stands. The mill is electrically operated throughout, with the main drive motors representing 42,250 hp, and an annual capacity of 600,000 tons.

Interpretation of Summer Weather Temperature Data for Design

By J. Donald Kroeker* and Harold V. Soballe**
Portland, Ore.

RECOMMENDED summer design dry- and wet-bulb temperatures, for use in the selection of proper heat exchange equipment suitable for air conditioning application, seem to vary widely according to various reported investigations. Despite careful individual study of probably identical local weather reports, these authorities indicate conclusions from recorded weather data which show wide variances.

Data collected from four available sources are shown in Table 1, giving recommended summer weather design data for a selected group of cities in the United States. Reference to the last column of these tabular data indicate differences as great as 17 and 9 F existing in the dry- and wet-bulb temperatures respectively. The lack of agreement of these design data may be due to the form of original record used, to the proper analysis of either simultaneous or non-simultaneous dry-bulb, wet-bulb or dew-point temperatures, to the disregard of a greater or lesser number of maximum conditions, because of rarity of occurrence, to method of analysis, etc.

A study[1] of summer weather design data was presented before the Society in 1939 which included tabular data of suggested design temperatures, iso-thermic maps of the United States showing dry-bulb, wet-bulb and dew-point design temperatures and other maps indicating summer prevailing winds and hours of sunshine. After making exhaustive studies of available records, the author concluded that more detailed weather information regarding individual cities should be collected by local interests, and that such efforts should be coordinated throughout the country.

The purpose of this paper is to indicate the various forms of local weather data which are available, to analyze these records, to study significant data which influence the selection of proper summer design conditions, to show variations in local conditions and to indicate means for appraising all of these various simultaneous factors. From such an analysis it is hoped that more complete information may eventually become available throughout the country, so that the designing engineer will have recommended criteria for design and thereby be able to more accurately adapt equipment for processes in which summer weather is a factor.

Weather Data Available

The U. S. Weather Bureau compiles data for specific purposes in two of its divisions and these include: (1) The Crop and Climate Division which records temperature data for 5,000 to 6,000 locations for use in connection with agricultural weather forecasts. (2) Aerological Division which records weather data for aviation purposes. Data compiled by the former division have generally been taken within cities and towns, and by the latter at airports or near cities.

The Crop and Climate Division obtains hourly weather data and summarizes selected data for all major cities in various publications, of which the *Monthly Meteorological Summaries* on Weather Bureau Form 1030 is of interest in this discussion. The information given on Form 1030 by days, consists of extreme temperatures, mean temperatures, dry-bulb temperatures and relative humidities or wet-bulb temperatures for at least three periods per day, as well as precipitation, wind, and sunshine data. Some stations also publish vapor pressures, river stage, and other useful local information.

The particular information furnished varies with the stations; and there is no uniformity among the stations in the times for which temperatures are recorded. For example, for a number of years prior to June, 1939, the Weather Bureau Station in Portland, Ore., recorded conditions for 4:30 a. m., local noon, and 4:30 p. m., and subsequent to that time, has published information for 4:30 a. m., 10:30 a. m., 4:30 p. m., and 10:30 p. m. Similarly, in 1940, data at Los Angeles, Calif., were published for 4:30 a. m., local noon, and 4:30 p. m.; at Denver, Colo., for 5:30 and 11:30 a. m., 5:30 and 11:30 p. m.; at Des Moines, Ia., for 12:30 and 6:30 a.m., 12:30 and 6:30 p. m.; for Cleveland, O., at 1:30 and 7:30 a. m., 1:30 and 7:30 p. m.; at Spokane, Wash., for 4:30 and 10:30 a. m., 4:30 and 10:30 p. m. Detailed records which show dry-bulb temperatures hourly and relative humidities or wet-bulb temperatures for three or four periods per day may be studied at the Weather Bureau Stations.

Data are obtained by the Aerological Division on Form 1140 and record hourly readings usually taken at airports. These records are not published, but are available for study at the airports or at the stations where they are taken. These records show monthly summaries of dry-bulb and dew-point temperatures. Some stations record hourly dew-point temperatures, and others record them at greater intervals.

In some cities the periodical atmospheric conditions shown in the published *Monthly Meteorological Summaries*, Form 1030, are now taken from data compiled by the Aerological Division and are indicated as *Airport Data*.

Method of Study

Selected available data were placed in graphic form, Figs. 1, 2, and 3, with dry-bulb and dew-point temperatures plotted against time, for the purpose of facilitating studies for selection of data source and of

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[1]Analysis of Summer Weather Data in the United States, by J. C. Albright. ASHVE TRANSACTIONS, Vol. 45, 1939, p. 397.)

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, San Francisco, Calif., June, 1941.

variation of temperature conditions within a locality, and to determine probable daily time of maxima. Records of simultaneous values of dry-bulb, wet-bulb, and dew-point temperatures were also plotted on a psychrometric chart,[2] Fig. 4, permitting rapid visual analysis of progression of atmospheric conditions. Selected extreme conditions pertinent to and affecting selection of summer design temperatures were plotted on similar subsequent charts to indicate records which may be preferable for study and to illustrate atmospheric conditions as they occur during a summer period in a number of selected cities in the United States.

Data for the various charts were taken from sources indicated on the charts or in the discussion.

Preferable Sources of Data

A comparison of data published by the Crop and Climate Division on Form 1030 and those recorded by the Aerological Division on Form 1140 for one week at Pittsburgh, Pa., is shown in Fig. 1. The data plotted from the former are maximum dry-bulb temperatures, and 8 a.m., local noon, and 8 p. m. dry-bulb and dew-point temperatures, the dew-point temperatures being converted from published relative humidities. Maximum dry-bulb temperatures are plotted in Fig. 1 at 4 p. m. daily, although information on actual time of occurrence is not shown on Form 1030. The data plotted from the Aerological Division data are dry-bulb and dew-point temperatures; the former from hourly records, the latter from records at 4 and 8 a. m., noon, 4 and 8 p. m., and midnight, all readings being taken 52 min after 75th Meridian time.

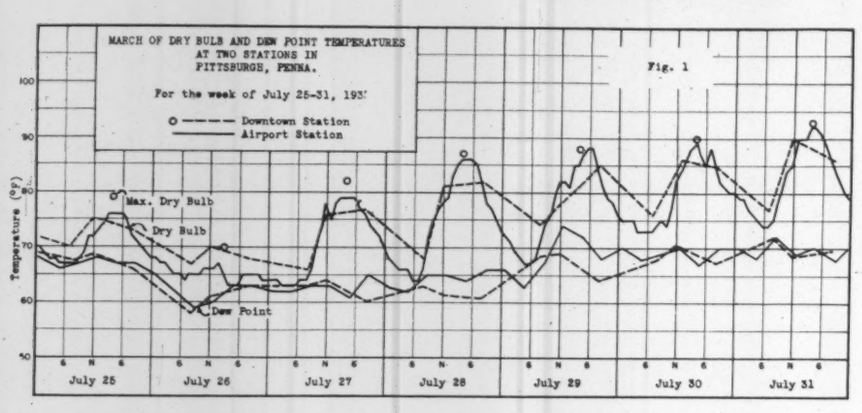
The maximum dry-bulb temperatures at Pittsburgh occur, according to the trend shown by the Airport records, between the hours of noon and 8 p. m. Therefore, the data published on Form 1030 for noon

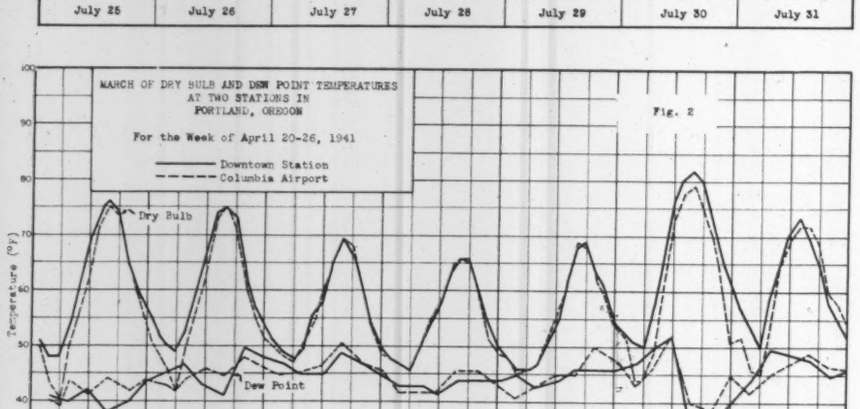
[2]Copyrighted form of psychrometric chart used by special permission.

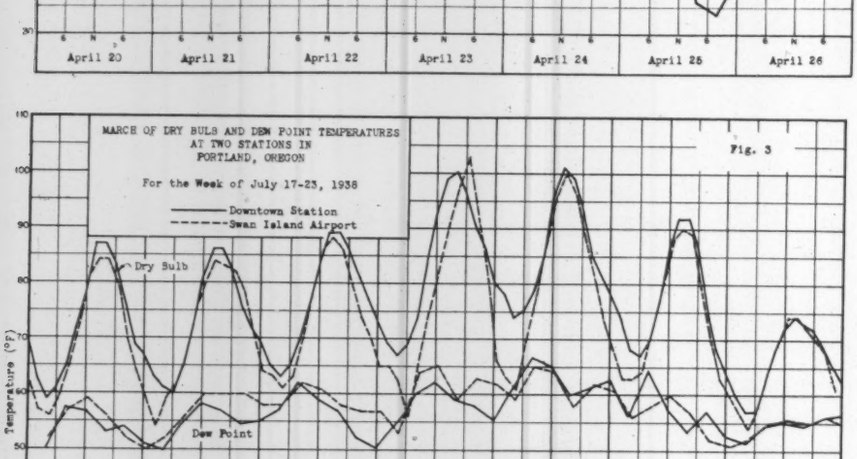
Table 1—Summer Design Temperatures According to Various Authorities[a] for Cities in the United States
(In Degrees Fahrenheit)

State	City	ASHVE		Albright		H. & V.		Mfr. A		Greatest Diff.	
		DB	WB	DB	WB	DB	WB	DB	WB	DB	WB
Alabama	Mobile	95	80	95	82	94	78	96	80	2	4
Arizona	Phoenix	105	76	113	78			102	76	11	2
Arkansas	Little Rock	95	78	103	83	95	77	96	79	8	6
California	Los Angeles	90	70			88	70	89	71	2	1
	San Francisco	90	65			85	68	84	65	6	3
Colorado	Denver	95	64	99	68			91	68	8	4
Connecticut	New Haven	95	75			88	74	91	74	7	1
Delaware	Wilmington	95	78					93	75	2	3
D. C.	Washington	95	78	99	84	95	78	94	77	5	7
Florida	Jacksonville	95	78	99	82	94	78	97	80	5	4
Georgia	Atlanta	95	76	101	82	91	75	95	77	10	7
Idaho	Boise	95	65	109	71			98	71	14	6
Illinois	Chicago	95	75	104	80	95	75	92	74	12	6
Indiana	Indianapolis	95	76			90	73	94	76	5	3
Iowa	Des Moines	95	77			92	74	93	75	3	3
Kansas	Wichita	100	75	110	79	95	77	95	77	15	4
Kentucky	Louisville	95	76			94	75	95	77	1	2
Louisiana	New Orleans	95	79	95	83	94	79	97	80	3	4
Maine	Portland	90	73					89	72	1	1
Maryland	Baltimore	95	78			93	76	94	77	2	2
Massachusetts	Boston	92	75	96	78	88	73	91	74	8	5
Michigan	Detroit	95	75	101	79	93	73	91	75	10	6
Minnesota	Minneapolis	95	75			84	72	93	72	11	3
Mississippi	Jackson			103	82			98	81	5	1
Missouri	Kansas City	100	76	109	79	92	75	94	76	17	4
Missouri	St. Louis	95	78	108	81	95	78	96	77	13	4
Montana	Helena	95	67	97	70			88	64	9	6
Nebraska	Omaha			108	80	94	75	94	75	14	5
Nevada	Reno	95	65	102	66					7	1
New Jersey	Trenton	95	78			95	76	90	72	5	6
New York	New York	95	75	100	81	95	75	92	75	8	6
	Albany	92	75	97	78	90	74			7	4
	Buffalo	93	75	93	77	83	72	90	73	10	5
New Mexico	Santa Fe	90	65					89	66	1	1
North Carolina	Wilmington	95	79	94	81			95	79	1	2
North Dakota	Bismarck	95	73			88	69	90	71	7	4
Ohio	Cincinnati	95	78	106	81	95	78	95	76	11	5
	Cleveland	95	75	101	79	95	73	92	74	9	6
Oklahoma	Oklahoma City	101	76			96	76	95	77	6	1
Oregon	Portland	90	65	99	70	83	65	88	68	16	5
Pennsylvania	Philadelphia	95	78			95	78	94	76	1	2
	Pittsburgh	95	75	98	79	91	73	93	75	7	6
Rhode Island	Providence	93	75			85	73			8	2
South Carolina	Charleston	95	80	98	82	94	80	97	80	4	2
South Dakota	Huron			106	76			91	73	15	3
Tennessee	Memphis	95	78	103	83			97	78	8	5
	Chattanooga	95	77			94	76			1	1
Texas	Dallas	100	78	105	80	99	76			6	4
	El Paso	100	69	101	72			96	73	5	4
	Houston	95	78	100	81	93	79			7	3
Texas	San Antonio	100	78	102	83	100	78			2	5
Utah	Salt Lake City	92	63	102	68	95	67	92	69	10	6
Vermont	Burlington	90	73			85	71	89	72	5	2
Virginia	Richmond	95	78			95	78			0	0
Washington	Seattle	85	65	86	70	83	61	86	67	3	9
Washington	Spokane	90	65	106	68	89	63			17	5
Wisconsin	Milwaukee	95	75			93	74	90	74	5	1
	Madison	95	75			89	73			6	2
Wyoming	Cheyenne	95	65								

[a]ASHVE, 1941 Guide; Albright, Loc. Cit. Note 7 p. 418; *Heating and Ventilating Magazine* reference data sheet No. 57, copyright 1935 by Industrial Press; Mfr. A, Handbook of one manufacturer's heating and air conditioning equipment.







and 8 p. m. do not indicate the maximum dry-bulb conditions. Also, the data on this form do not indicate information from which dew-point may be computed between the hours of noon and 8 p. m. Therefore, the data published on Form 1030 are not suitable for a study towards selection of design conditions consisting of dry-bulb temperatures and dew-point or wet-bulb temperatures. In order to show the temperature occurring at the downtown station for studies in this connection, it would be necessary to study the detailed office records on file at the Pittsburgh station for the downtown station.

A number of the published reports of the Crop and Climate Division of the Weather Bureau in other cities do not show weather data for mid-afternoon periods during which maximum design conditions usually occur, as is shown in later discussions. It is therefore unsatisfactory to rely upon the published data for studies in this connection, unless these data are given for mid-afternoon periods.

Variation Due to Locality Within a City

A comparison of dry-bulb and dew-point temperatures taken simultaneously at the Weather Bureau Station established in downtown Portland, and those taken at the Swan Island Airport, approximately 3 miles from the downtown station, is shown in Fig. 2. Dry-bulb temperatures are plotted for alternate hours and dew-point temperatures for every fourth hour. The Airport is on an island in the Willamette River and is surrounded by water; the downtown station is approximately one-fourth mile from the Willamette River, which parallels the prevailing summer wind direction. A wide variation both in dry-bulb and dew-point temperatures at the two stations will be observed.

In general the downtown weather recording station shows a higher dry-bulb temperature and a typical progression of dew-point temperatures, with a trend at times entirely unrelated to that recorded at the airport. Dew-point temperatures at the airport are consistently higher. Such a variation is sufficient to affect a normal selection of design dry-bulb and wet-bulb temperatures for a particular application in this locality.

A similar comparison of temperature records for the downtown weather station and those for the Columbia Airport, 6 miles distant and adjacent to the Columbia River, which is in the transverse path of prevailing summer winds, is shown in Fig. 3. This station has been recently established and records are available for only about six months. However, it is evident from comparison of one week's temperature records at the downtown station and at Columbia Airport that maximum dry-bulb temperatures at the latter are well below those of the downtown station, and the diurnal range is greater. The dew-point curves have dissimilar trends throughout, as is shown for the morning and afternoon of April 20, the afternoon of April 21, the early morning of April 24, and the afternoon of April 24. It appears from this fragmentary record that the dew-point temperatures during maximum dry-bulb temperature will

be higher at the Columbia Airport than at the downtown station.

It is apparent from Figs. 2 and 3 that records of the dry-bulb and dew-point temperatures obtained at the Swan Island and Columbia Airports, if used as bases for design, will probably result in selection of lower dry-bulb and higher dew-point design temperatures than justified by study of simultaneous records taken at the downtown station.

Data for two stations at Pittsburgh, Fig. 1, also show marked variations in conditions between two recording stations in the same city, although, for the week charted, close agreement is found on days of extreme temperatures.

These comparisons indicate that an error may result in use of the temperature data obtained in one part of a large city for application in another part of the same city, and that variation of atmospheric conditions and consequent error may be considerable across any one city. This should be particularly true for cities located near large bodies of water, where the prevailing wind direction is such that only a part of the locality is in the path of the winds passing over the water surfaces.

The general Portland area includes Vancouver, Wash., which is north and on the windward side of the Columbia River and approximately 7 miles from the downtown Portland weather station. Weather data are not available for Vancouver, but in view of its location, it should be expected that dew-point temperatures should be considerably lower than those recorded at the Portland downtown station or at either of the airport stations.

Experience has indicated some wide variation between conditions at locations at which the Weather Bureau records are taken and those at which the engineer must determine heat transfer to buildings or plan to introduce and condition air for ventilating purposes. Further studies of variations of conditions within a locality should be made to establish relationships to Weather Bureau data and to determine desirable corrections for such data.

Time of Record

From Figs. 2 and 3, it is apparent that maximum dry-bulb temperatures for the Portland data studied occurred generally between 2 and 4:30 p. m., and that, while there was, generally, some decline in dew-point at this period, the wet-bulb was near the maximum during this period. From these illustrations of daily temperature progressions, it appears that atmospheric conditions for the daily period between 2 and 4:30 p. m. are satisfactory for the purpose of establishing design bases.

The wet-bulb temperature is of as great importance and interest in connection with design temperature selection as the dew-point temperature. This factor may be studied simultaneously with dry-bulb and dew-point temperatures in Fig. 4, which shows the progression of atmospheric conditions for three days, beginning July 19, 1938, at Portland, and is typical for this city. The general daily trend is an increase of the total heat between the hours of 4 and 8 a. m.; a continuing increase from 8 a. m. to noon; fairly constant wet-bulb between noon and 4 p. m.; and subsequently, a constantly diminishing trend.

For study of periodic atmospheric conditions extending over greater time intervals, Fig. 5 shows four conditions daily for the entire year of 1940 as recorded at Portland; Fig. 6, three daily for the summer of 1940; and Fig. 7, two daily for the same period.

Generally, for Portland, no additional information is contributed by use of atmospheric data on or for the periods other than that of 4:30 p. m. This is true also for Pendleton, Ore., as indicated by study of Figs. 8 and 9. The only significant fact disclosed in these five charts, Figs. 5 to 9, is that the concentration of conditions moves to the left and downward on the chart, so that conditions other than those for mid-afternoon are eliminated from useful consideration in selection of a design condition.

Years of Record

The atmospheric conditions indicated on the previous psychrometric charts, Figs. 4 to 9, are for individual years. It is obviously undesirable to design on a basis of data obtained from weather data for a single year. Four years of atmospheric conditions at Portland during June, July, August, and September occurring at 4:30 p. m. during the years 1937, 1938, 1939, and 1940 are shown in Fig. 10. This composite chart covering the four years includes a number of points indicating atmospheric conditions which did not occur in the year 1940, shown on Figs. 5 to 9. In order to select a suitable design temperature, it would be desirable to show atmospheric conditions as in Fig. 10 for at least five years, and preferably 10 years. Fig. 10 shows also the design temperature recommended for Portland by the 1941 ASHVE GUIDE[3] and the previously mentioned paper.[4]

Using information of atmospheric conditions in the manner presented permits the designing engineer to make personal selections of design temperature, with reference to both the number of conditions that are deemed inadvisable to design for, and for the selection of proper cooling equipment.

If a conclusion as to design data for Portland were to be developed from the data shown on Fig. 10, a dry-bulb temperature of 95 F, a wet-bulb temperature of 70 F, and a dew-point temperature of 63 F would be logical maximum design conditions on the basis that the wet-bulb temperature, the dry-bulb temperature, and absolute humidities at these conditions are exceeded separately on not more than eight days in the four years analyzed. Such design conditions would then lie to the left of and below the heavy lines shown in Fig. 10. Conditions lying outside of this area would be ignored as being of occurrence so rare as not to warrant consideration in comfort air conditioning applications.

Studies of Other Cities

A number of psychrometric charts are presented in Figs. 11 to 17 to show summer atmospheric conditions for cities other than those previously analyzed. These charts are not suitable for selection of summer design conditions, since they are for a single year. They are presented to show at a glance the probably typical variation existing between cities and some immediate conclusion possible as to design of equipment.

The psychrometric chart for Spokane, Fig. 11, indicates that all summer conditions for the year shown

[3] HEATING, VENTILATING, AIR CONDITIONING GUIDE, 1941, p. 126.
[4] Loc. Cit. Note 1, p. 148.

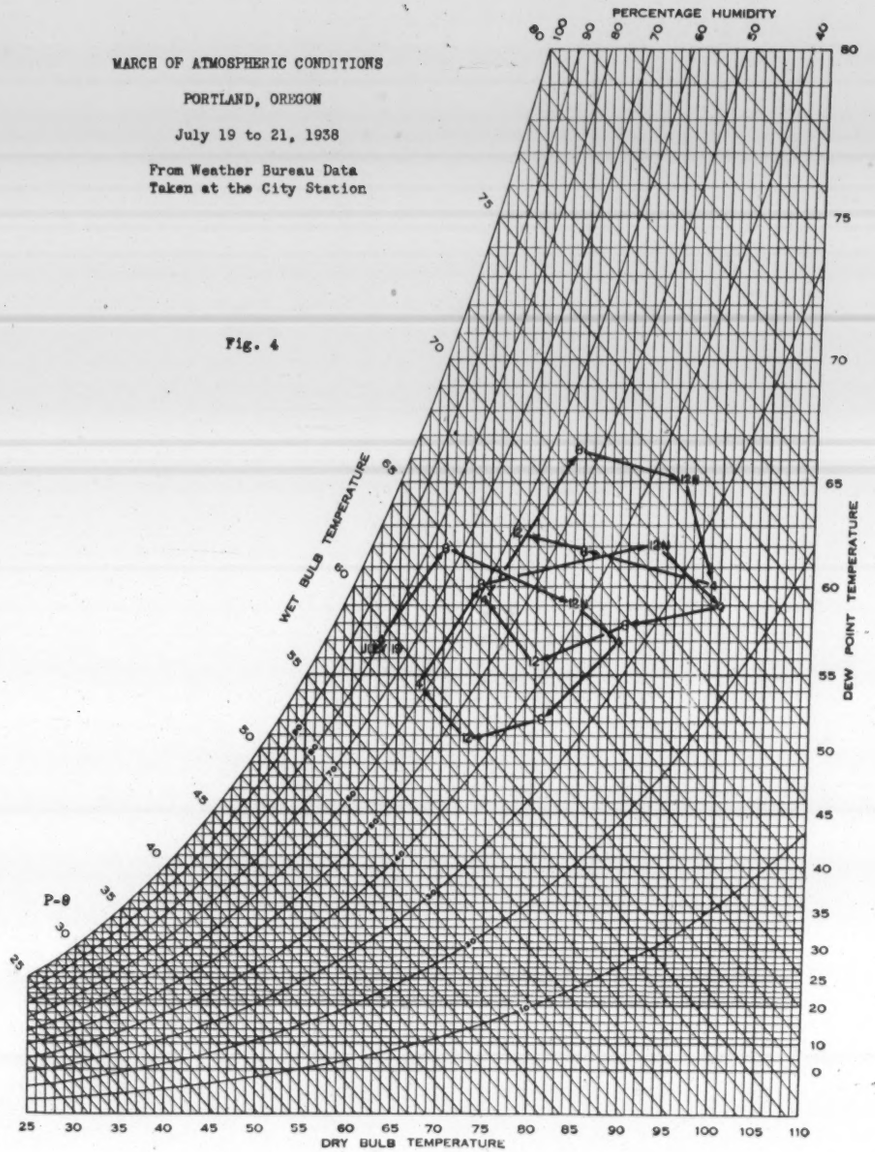
MARCH OF ATMOSPHERIC CONDITIONS

PORTLAND, OREGON

July 19 to 21, 1938

From Weather Bureau Data
Taken at the City Station

Fig. 4



ATMOSPHERIC CONDITIONS

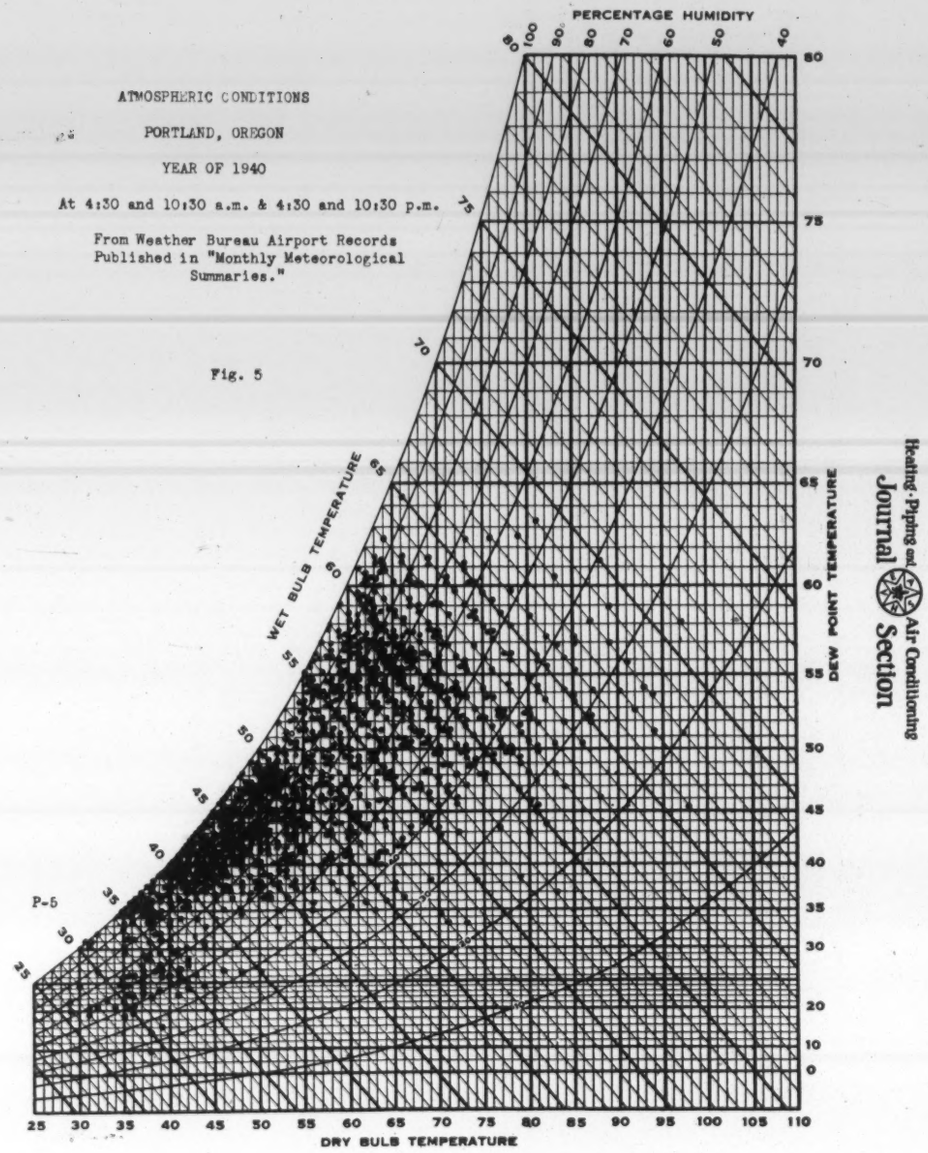
PORTLAND, OREGON

YEAR OF 1940

At 4:30 and 10:30 a.m. & 4:30 and 10:30 p.m.

From Weather Bureau Airport Records
Published in "Monthly Meteorological Summaries."

Fig. 5



Heating · Piping and Air Conditioning Journal Section

ATMOSPHERIC CONDITIONS

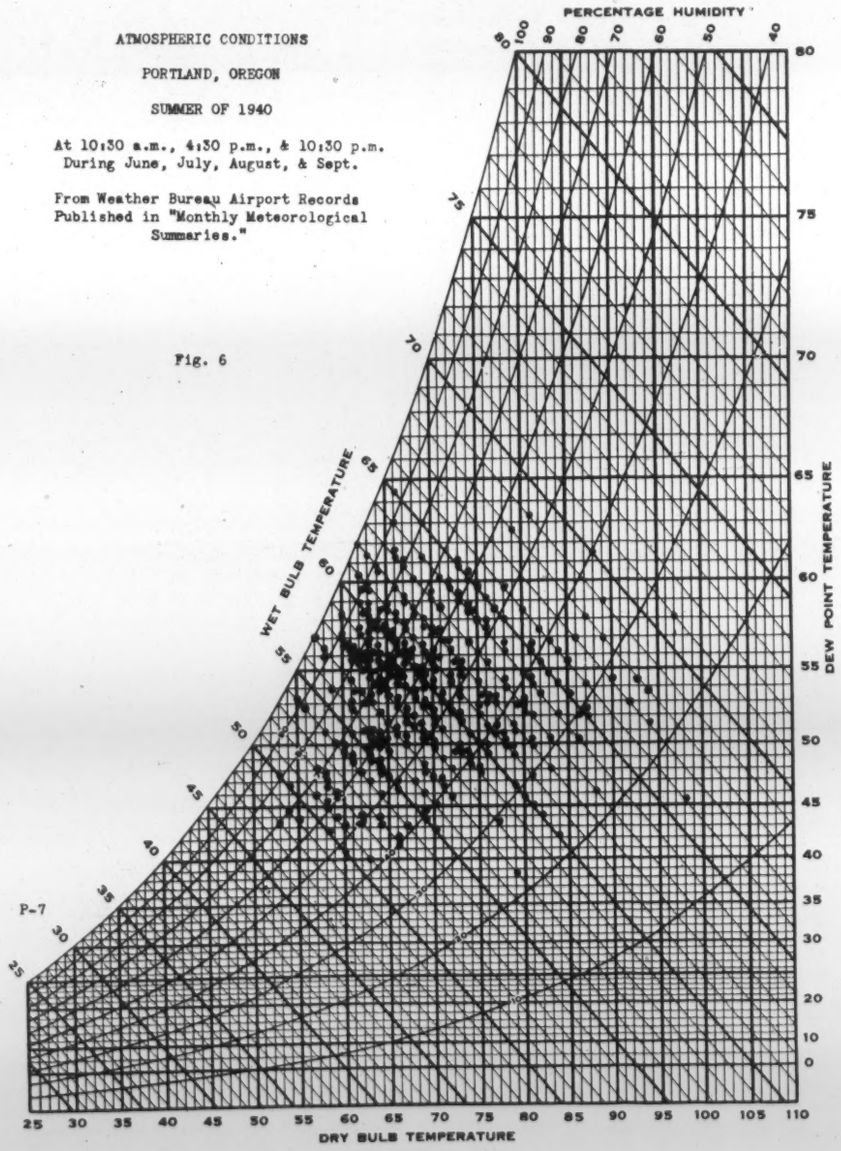
PORTLAND, OREGON

SUMMER OF 1940

At 10:30 a.m., 4:30 p.m., & 10:30 p.m.
During June, July, August, & Sept.

From Weather Bureau Airport Records
Published in "Monthly Meteorological Summaries."

Fig. 6



ATMOSPHERIC CONDITIONS

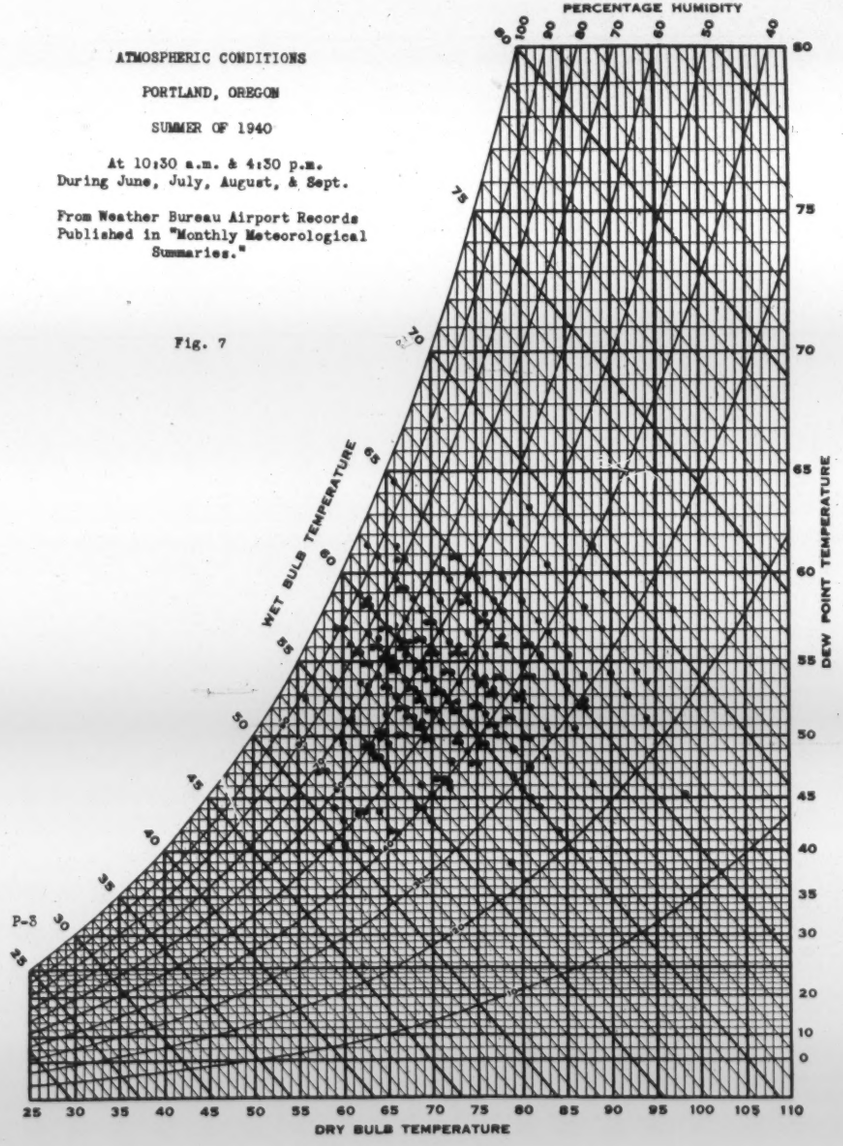
PORTLAND, OREGON

SUMMER OF 1940

At 10:30 a.m. & 4:30 p.m.
During June, July, August, & Sept.

From Weather Bureau Airport Records
Published in "Monthly Meteorological Summaries."

Fig. 7



ATMOSPHERIC CONDITIONS

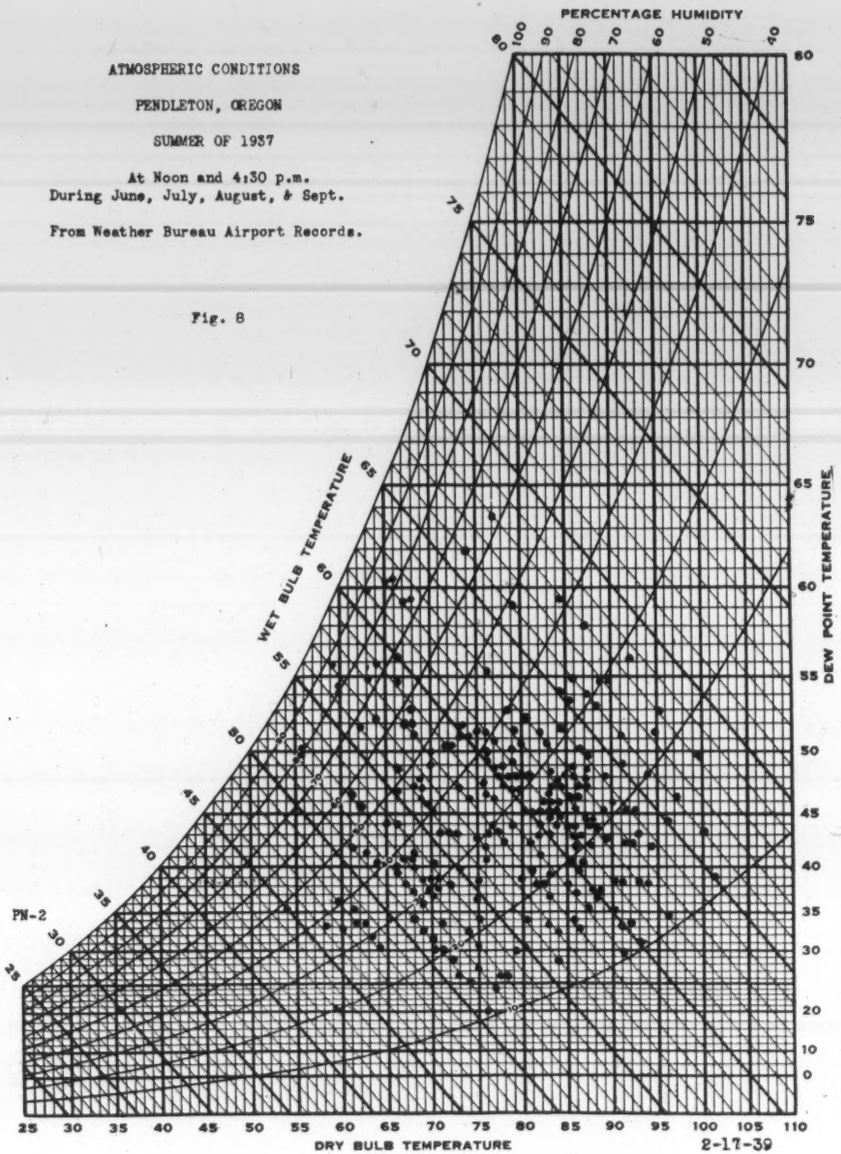
PENDLETON, OREGON

SUMMER OF 1937

At Noon and 4:30 p.m.
During June, July, August, & Sept.

From Weather Bureau Airport Records.

Fig. 8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100 90 80 70 60 50 40
WET BULB TEMPERATURE
DEW POINT TEMPERATURE
DRY BULB TEMPERATURE
25 30 35 40 45 50 55 60 65 70 75 80 85 90 95 100 105 110
PN-2
2-17-39

ATMOSPHERIC CONDITIONS

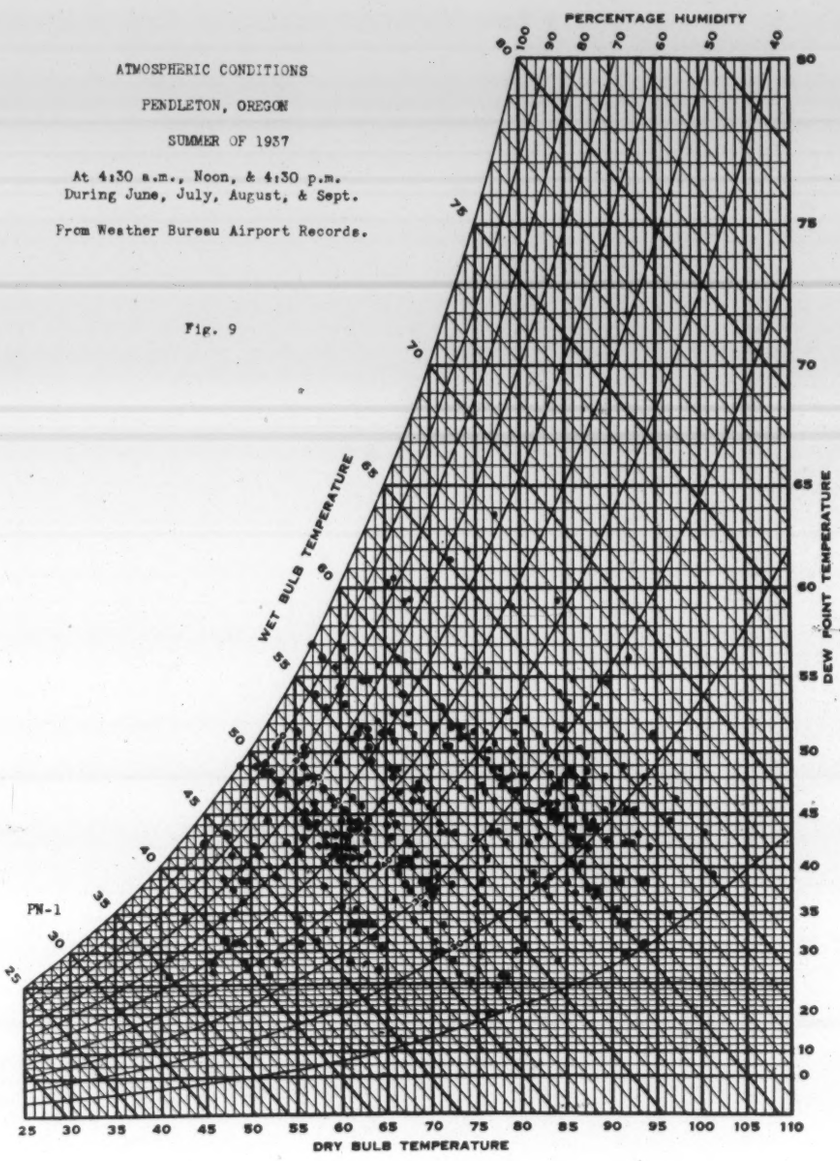
PENDLETON, OREGON

SUMMER OF 1937

At 4:30 a.m., Noon, & 4:30 p.m.
During June, July, August, & Sept.

From Weather Bureau Airport Records.

Fig. 9


PERCENTAGE HUMIDITY
100 90 80 70 60 50 40
WET BULB TEMPERATURE
DEW POINT TEMPERATURE
DRY BULB TEMPERATURE
25 30 35 40 45 50 55 60 65 70 75 80 85 90 95 100 105 110
PN-1

Heating · Piping and Air Conditioning Journal Section

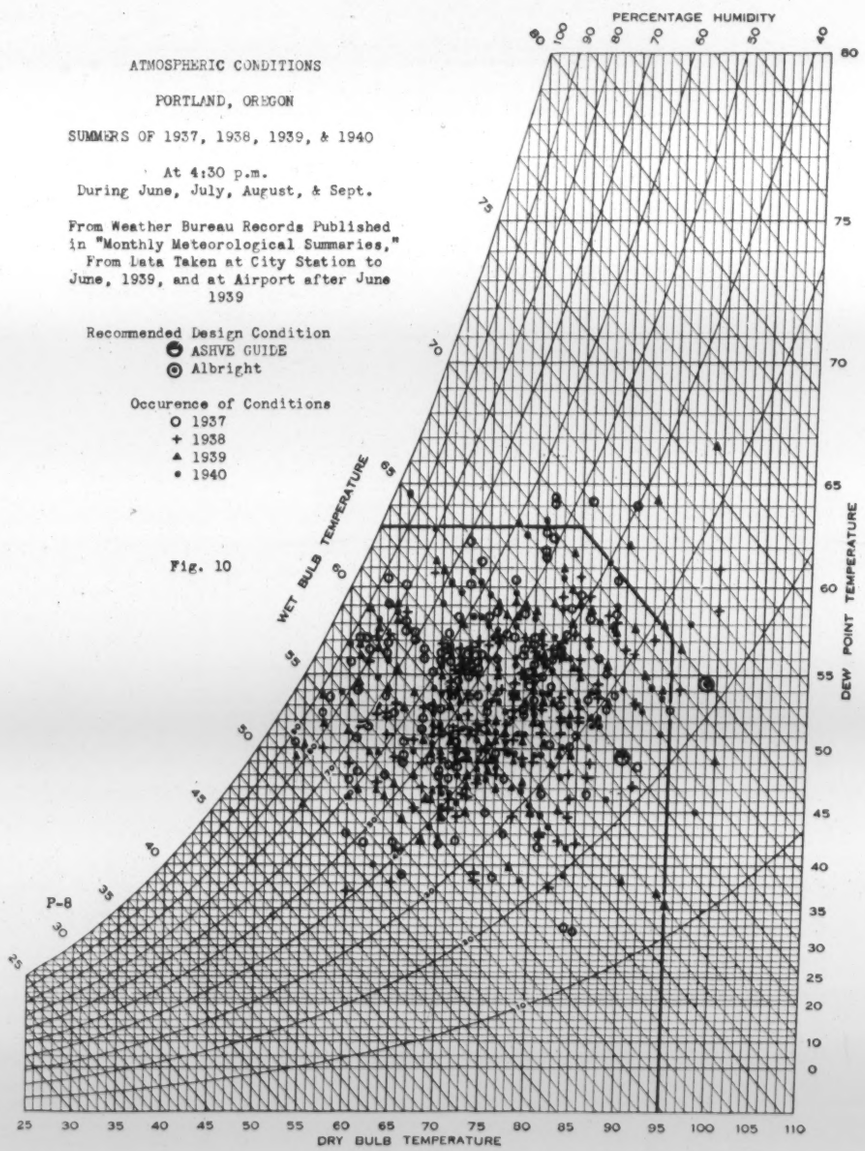

ATMOSPHERIC CONDITIONS
PORTLAND, OREGON
SUMMERS OF 1937, 1938, 1939, & 1940
At 4:30 p.m.
During June, July, August, & Sept.
From Weather Bureau Records Published in "Monthly Meteorological Summaries," From Data Taken at City Station to June, 1939, and at Airport after June 1939
Recommended Design Condition
ASHVE GUIDE
Albright
Occurence of Conditions
1937
1938
1939
1940
PERCENTAGE HUMIDITY
WET BULB TEMPERATURE
DEW POINT TEMPERATURE
DRY BULB TEMPERATURE
P-8

Fig. 10

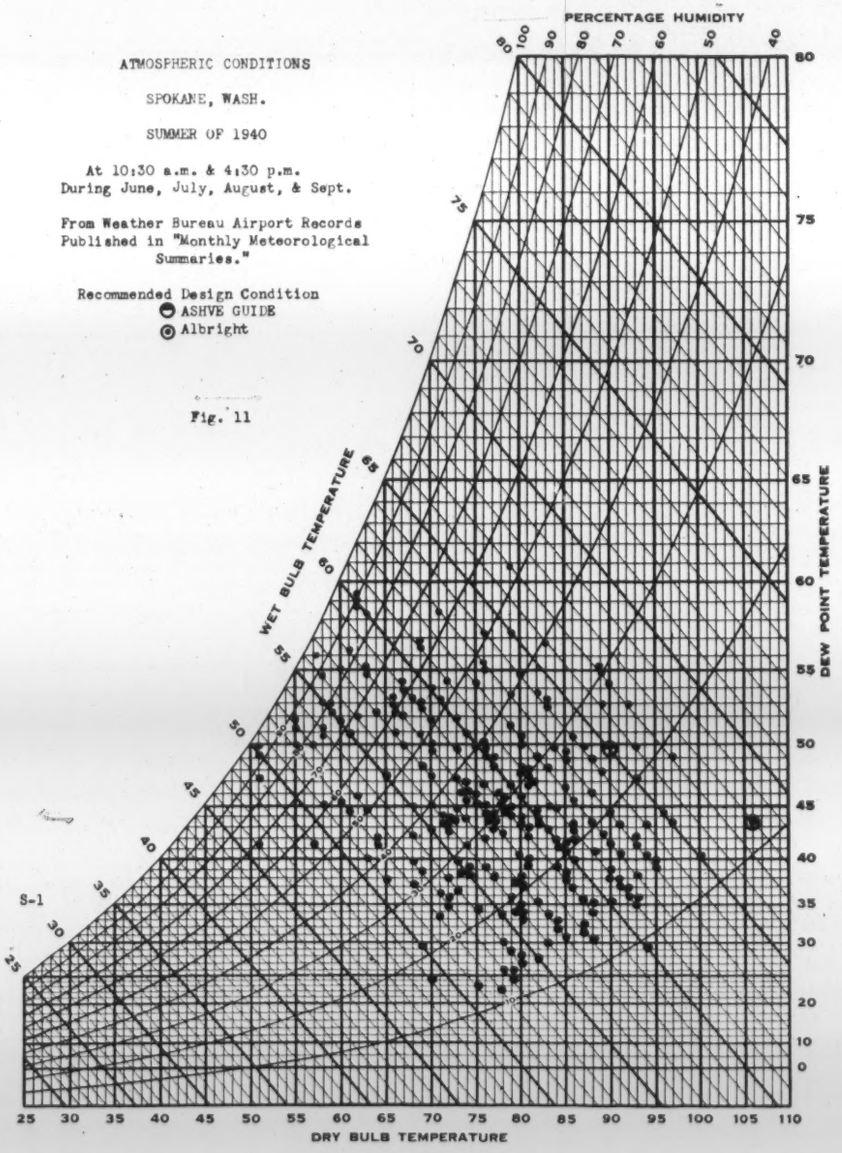

ATMOSPHERIC CONDITIONS
SPOKANE, WASH.
SUMMER OF 1940
At 10:30 a.m. & 4:30 p.m.
During June, July, August, & Sept.
From Weather Bureau Airport Records Published in "Monthly Meteorological Summaries."
Recommended Design Condition
ASHVE GUIDE
Albright
PERCENTAGE HUMIDITY
WET BULB TEMPERATURE
DEW POINT TEMPERATURE
DRY BULB TEMPERATURE
S-1

Fig. 11

ATMOSPHERIC CONDITIONS

DES MOINES, IOWA

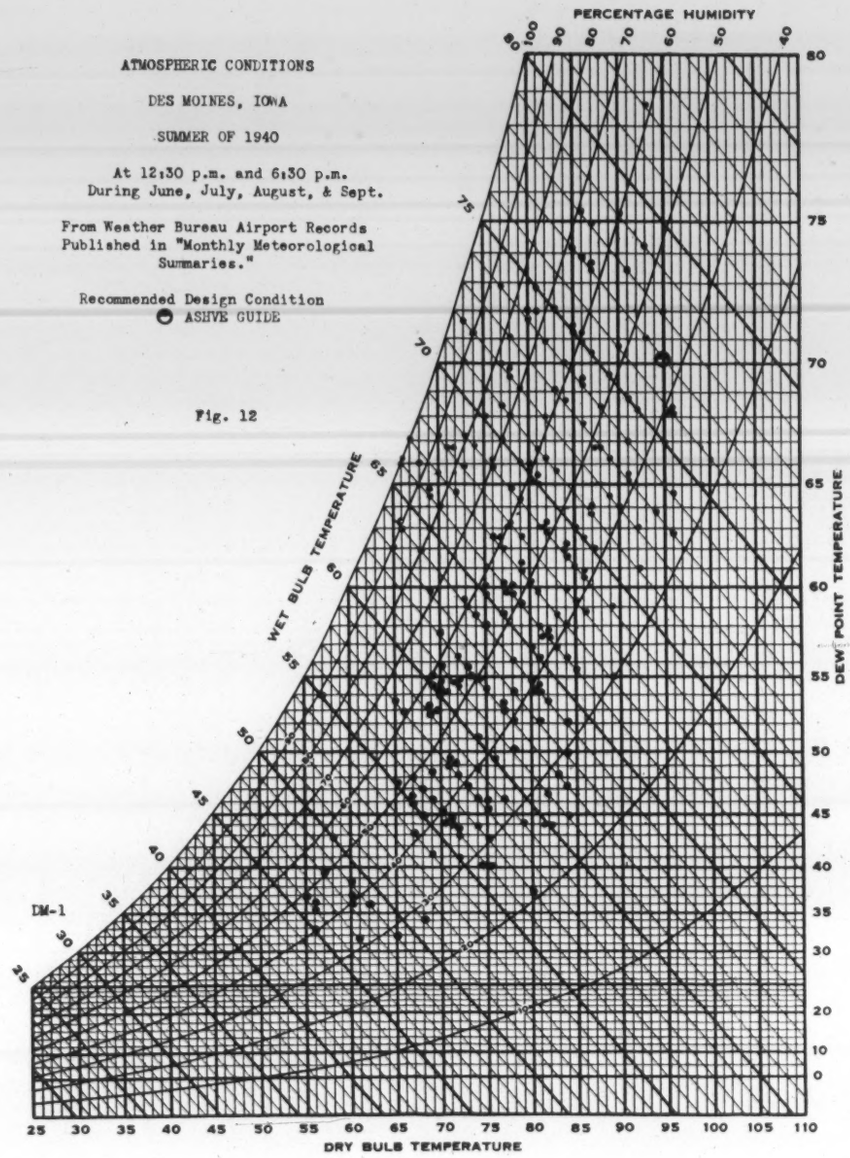
SUMMER OF 1940

At 12:30 p.m. and 6:30 p.m.
During June, July, August, & Sept.

From Weather Bureau Airport Records
Published in "Monthly Meteorological
Summaries."

Recommended Design Condition
⊖ ASHVE GUIDE

Fig. 12



ATMOSPHERIC CONDITIONS

LOS ANGELES, CALIFORNIA

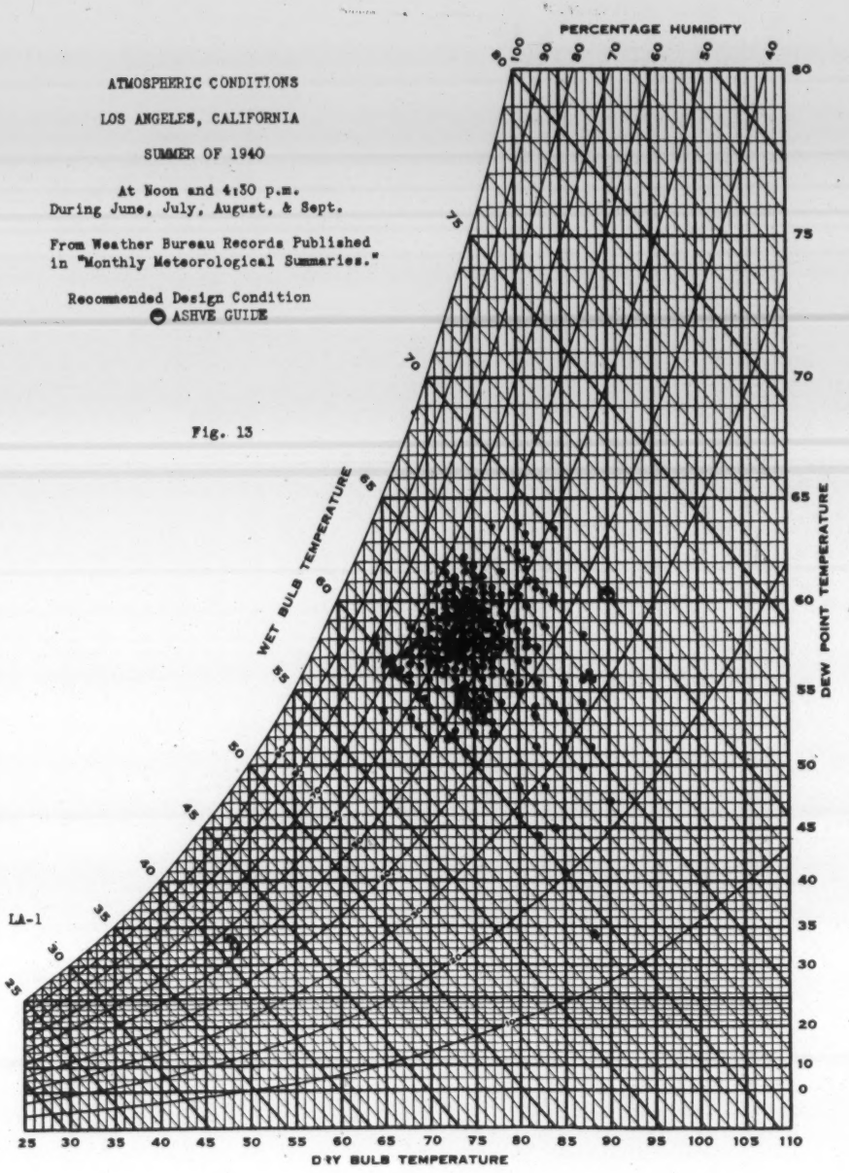
SUMMER OF 1940

At Noon and 4:30 p.m.
During June, July, August, & Sept.

From Weather Bureau Records Published
in "Monthly Meteorological Summaries."

Recommended Design Condition
⊖ ASHVE GUIDE

Fig. 13



Heating · Piping and Air Conditioning Journal Section

ATMOSPHERIC CONDITIONS

DENVER, COLORADO

SUMMER OF 1940

At 11:30 a.m. & 5:30 p.m.
During June, July, August, & Sept.

From Weather Bureau Airport Records
Published in "Monthly Meteorological Summaries."

Recommended Design Condition
- ASHVE GUIDE
- Albright

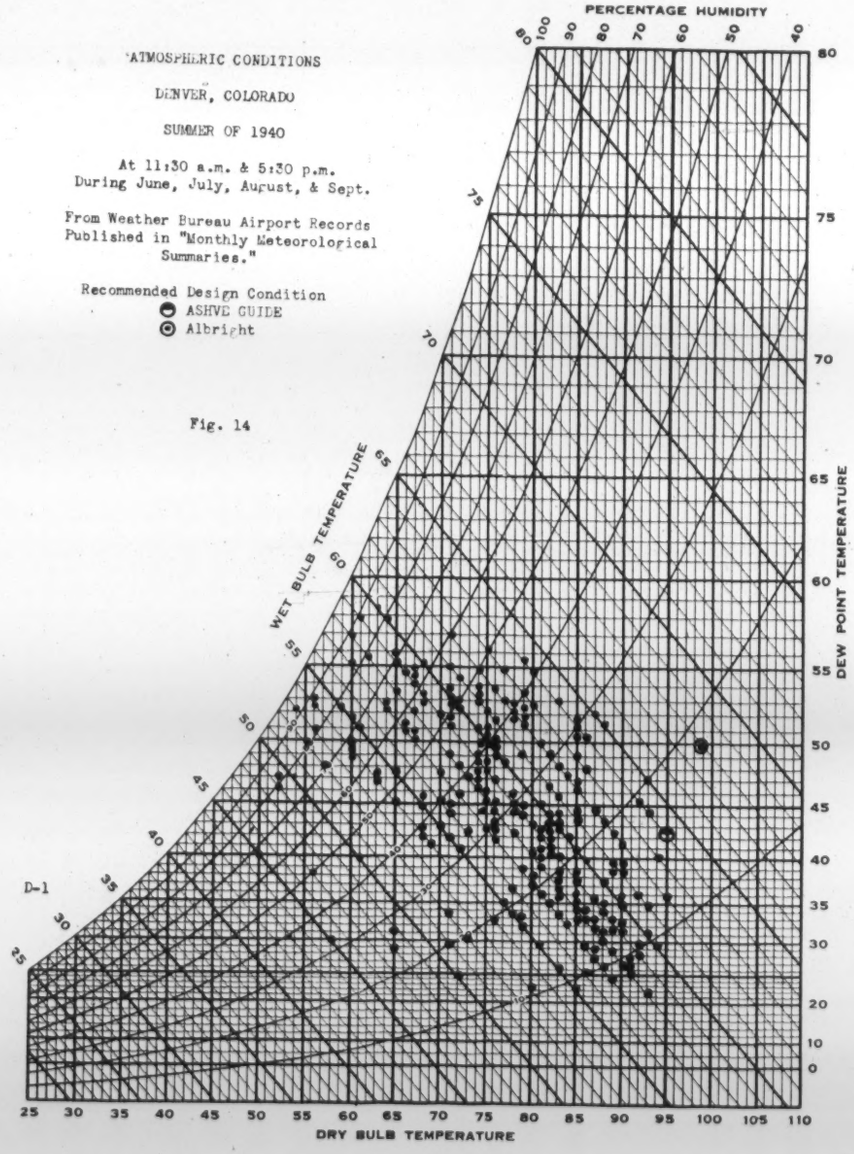


Fig. 14

ATMOSPHERIC CONDITIONS

NEW ORLEANS, LOUISIANA

SUMMER OF 1940

At 12:30 and 6:30 p.m.
During June, July, August, & Sept.

From Weather Bureau Airport Records
Published in "Monthly Meteorological Summaries."

Recommended Design Condition
- ASHVE GUIDE
- Albright

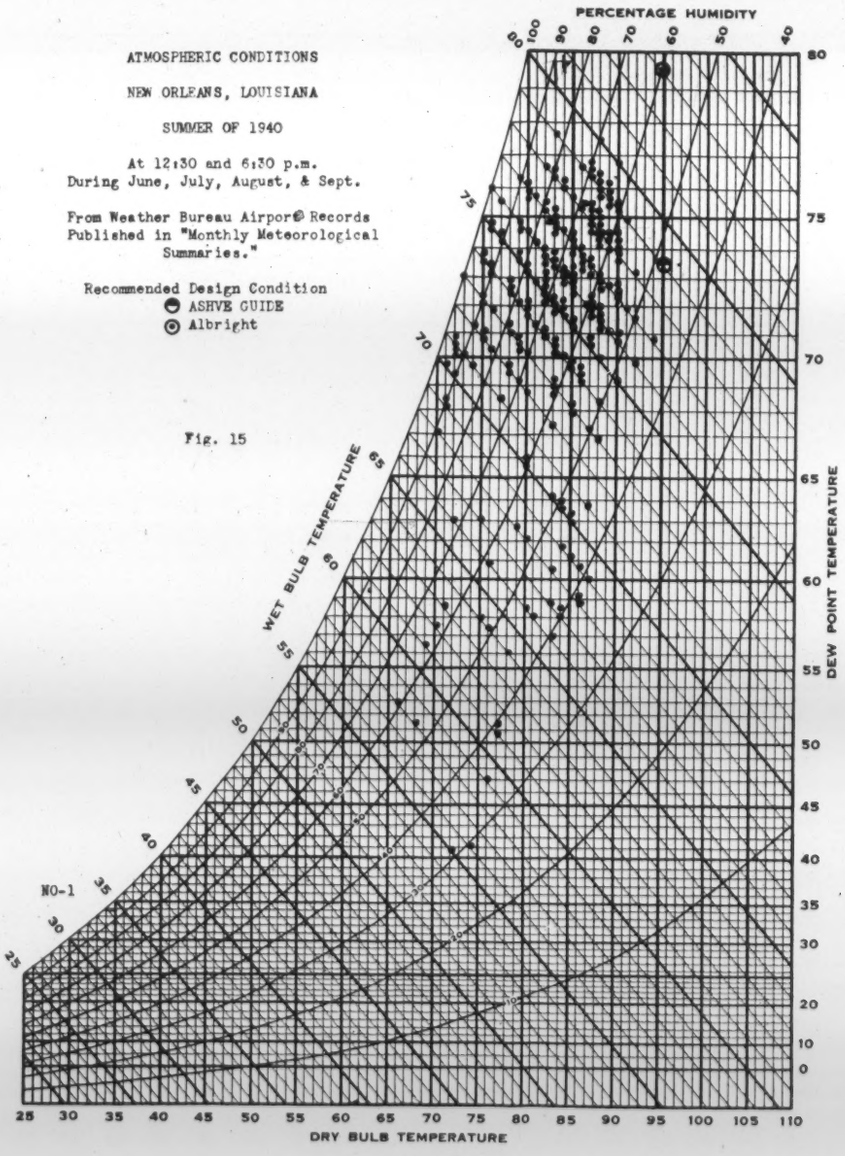


Fig. 15

ATMOSPHERIC CONDITIONS

CLEVELAND, OHIO

SUMMER OF 1940

At 1:30 p.m.
During June, July, August, & Sept.

From Weather Bureau Airport Records
Published in "Monthly Meteorological Summaries."

Recommended Design Condition
- ASHVE GUIDE
- Albright

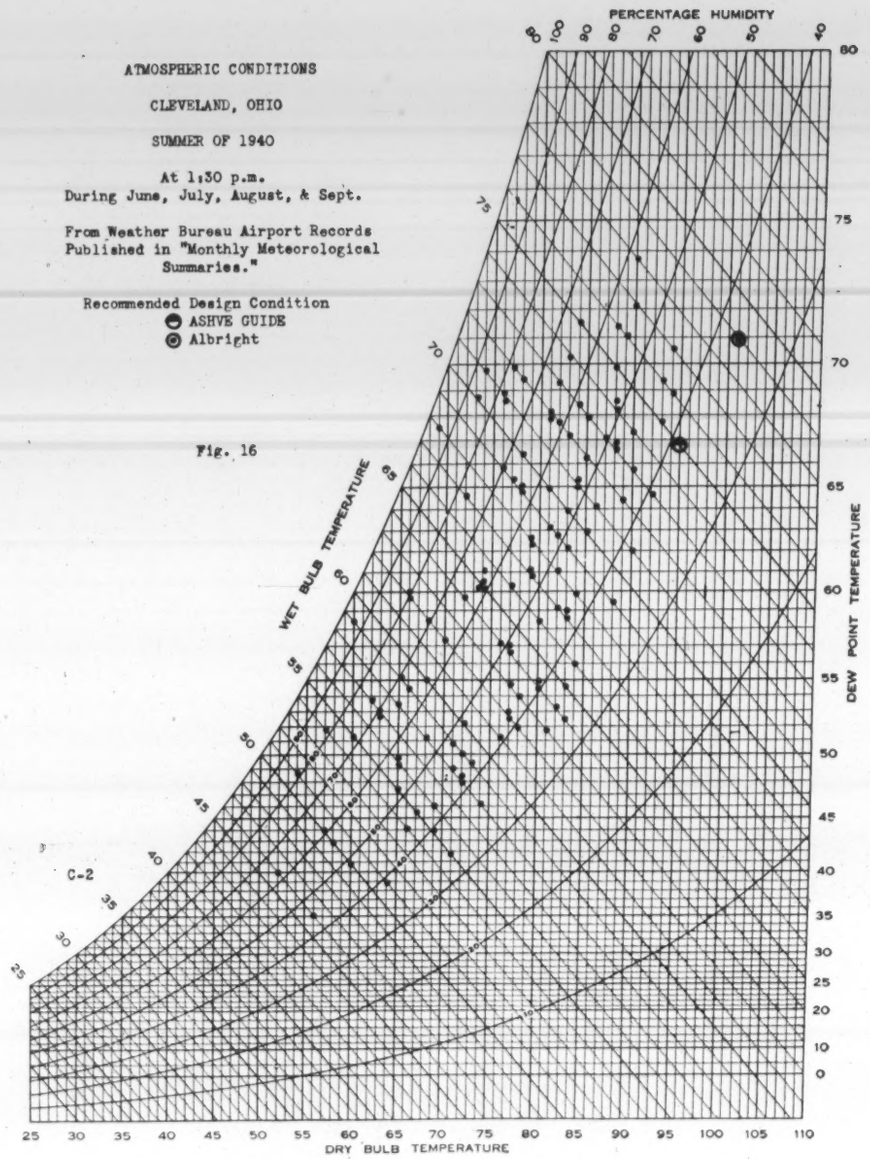


Fig. 16

ATMOSPHERIC CONDITIONS

MEDFORD, OREGON

SUMMER OF 1938

At 4:30 p.m.
During June, July, August, & Sept.

From Weather Bureau Airport Records.

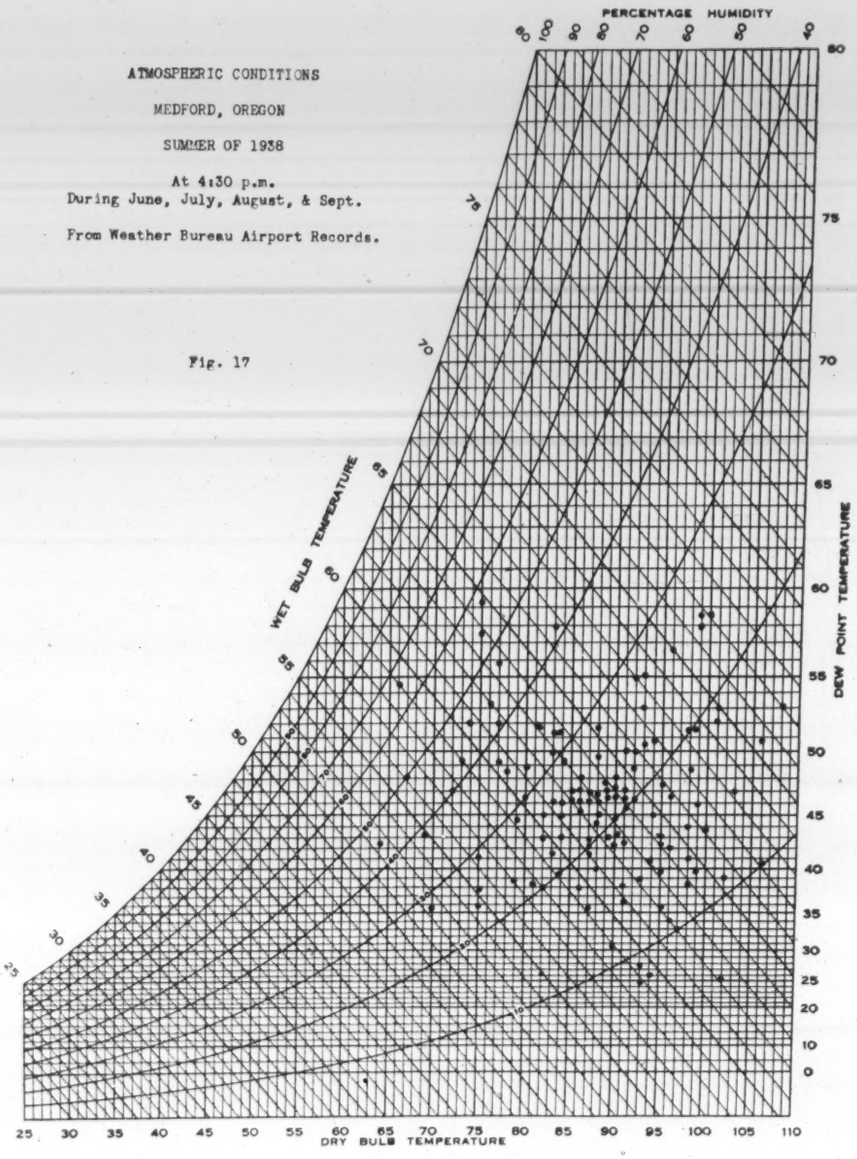


Fig. 17

Heating · Piping and Air Conditioning Journal Section

occurred at comparatively low dew-points. It would, therefore, be logical to conclude that evaporative cooling could be employed satisfactorily under certain conditions in installations using large proportions of outdoor air in this area.

The chart of atmospheric conditions for Des Moines, Fig. 12, indicates that evaporative cooling would not be adaptable; that maximum cooling loads occur at irregular intervals; and that compressor capacity for maximum daily loads could be divided among several units, so that, for example, a single compressor capacity might carry the cooling load for conditions below 70 F wet-bulb, two compressors between 75 and 70 F wet-bulb, and that a third compressor would be employed at conditions above the 75 F wet-bulb line. The chart for Des Moines is not suitable for selection of design data since only 12:30 and 6:30 p. m. data are shown. The maxima probably occurring in mid-afternoon are not recorded, no data being given for this period in the published Form 1030 of the Weather Bureau.

Fig. 13, showing atmospheric conditions for Los Angeles, would indicate the inadvisability of evaporative cooling, and that full compressor capacity might be used daily throughout the season, with a probability that, under usual air conditioning practices, it might be inadvisable to stagger compressor loads.

Fig. 14 shows the atmospheric conditions occurring at Denver, Colo., during 1940. Here again there might be some room for the application of the evaporative cooling. Dew-points are nearly all below the 55 F line, so that cooling with a fairly high temperature medium might be fully satisfactory, and well water or city water might be fully adaptable. Mid-afternoon peaks are not recorded.

Atmospheric conditions occurring in New Orleans during 1940 are indicated in Fig. 15. The requirements, as indicated by the chart, are for a compressor load daily at nearly full capacity throughout the summer. While afternoon readings are for 6:30 p. m., no great variation will be found for 1940 from the conditions shown, due to the limited diurnal range in New Orleans, which is shown by study of other Weather Bureau data.

Fig. 16, a psychrometric chart of atmospheric conditions for Cleveland, would indicate that considerable variations in atmospheric conditions occur during the summer season; and that in view of the high dew-point temperatures prevalent during a greater portion of the time, evaporative cooling would be out of the question. The atmospheric conditions plotted are for readings at 1:30 p. m. and should be satisfactory for studies toward the development of required design temperatures, since the summer diurnal range is low and the conditions at three and four o'clock should not be much more severe than those shown.

The summer atmospheric conditions for Medford, Ore., at 4:30 p. m. are shown in Fig. 17. The conditions at Medford are at the opposite extreme from those at New Orleans, shown on Fig. 15. The dew-point temperature is low, making cooling with relatively high temperature medium or evaporative cooling feasible under favorable conditions.

From Figs. 11 to 17, it may be further concluded that the number of atmospheric conditions occurring annually, which may be ignored in selection of a summer design condition, will vary with the climate peculiar to each location. Summer conditions for Los Angeles and New Orleans, for example, are closely grouped, minor departures from conditions of usual occurrence may be expected frequently, and fewer conditions may be omitted from consideration. Those for Des Moines, Denver, Cleveland, Spokane, and Medford, however, are not closely grouped and single departures, as shown inconclusively in the single-year studies, are few but of considerable magnitude. Failure to use these departures in establishing the design datum for each of these cities may result in unsatisfactory indoor conditions for at most a day or two per year, which may or may not be justified as against greater first cost of equipment and possible greater power demand and other charges. However, with weather data of at least five years available for analyzation in the form of psychrometric charts, the designing engineer may study the acceptable medium for each installation.

Use of Data in Design

Psychrometric charts with atmospheric conditions plotted as shown have been found very convenient forms in studying equipment design and function. For this purpose, the charts are printed in an ozalid or similar process so that the prints and the data shown on them can form a background for the diagrammatic study necessary in design of equipment.

Psychrometric charts similar to those used in this paper and other forms developed by other individuals are available in pad form and, when properly transparentized, will print clearly.

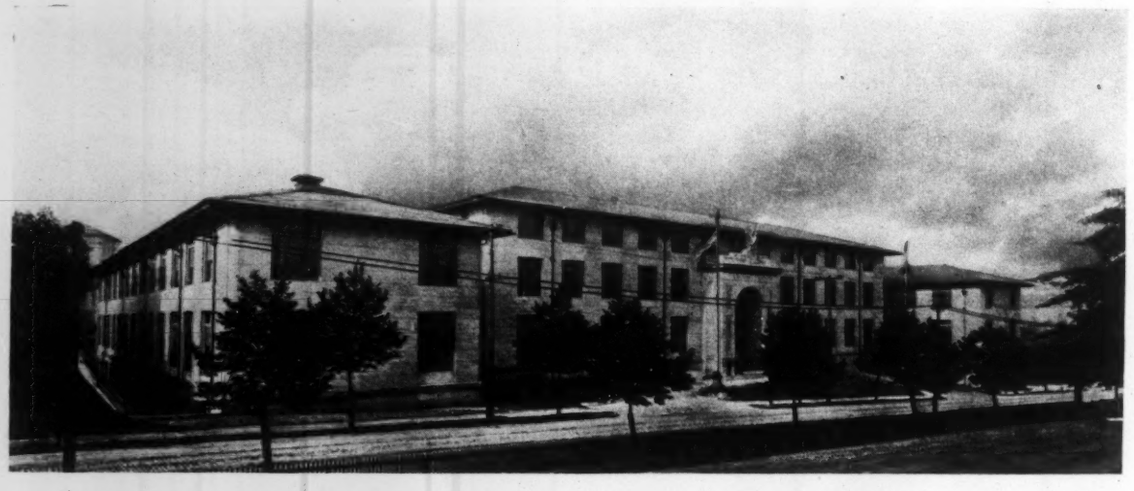
Conclusions

Data compiled by the Aerological Division and detailed office records of the Crop and Climate Division of the U. S. Weather Bureau are more complete for the purpose of studies in a development of satisfactory summer design temperature criteria than those published by the Crop and Climate Division. Atmospheric conditions may vary widely in any one locality, and data obtained at a weather recording station are not necessarily fully applicable at locations as little as 5 miles distant.

For purposes of establishing summer design conditions, daily weather records for conditions between 2 and 4:30 p. m. should be used; design conditions based on earlier or later temperatures may be misleading, unless it is found that the diurnal range is low.

Psychrometric charts lend themselves satisfactorily to study of design conditions, offering a means of plotting dry-bulb, wet-bulb, and dew-point temperatures simultaneously. When so used, they provide a satisfactory basis of analysis of air conditioning problems with all pertinent temperature data as a background.

Further studies of variations in weather conditions within a locality should be made, with a view to establishing possible relationships between proximity to open water surfaces, wind directions, concentrated city areas, conditions at street levels, etc. Further local studies of atmospheric conditions should be instituted to establish design data, which should be made locally and assembled nationally for dissemination through sources and authorities relied upon for such information.



Pittsburgh Experiment Station of the U. S. Bureau of Mines where the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS is located

Local Cooling of Workers in Hot Industry

By F. C. Houghten*, D.Sc., M. B. Ferderber**, M.D., and Carl Gutberlet†
Pittsburgh, Pa.

THIS paper gives the results of studies involving physiological reactions of persons to hot atmospheres which were carried on at the ASHVE Research Laboratory, U. S. Bureau of Mines, Pittsburgh, Pa., under the Technical Advisory Committee on Air Conditioning in Industry, consisting of: W. L. Fleisher, *Chairman*; Philip Drinker, Dr. Leonard Greenburg, H. P. Greenwald, A. M. Kinney, J. W. Kreuttner, L. L. Lewis, H. B. Matzen, Dr. W. J. McConnell, Dr. C. P. McCord, P. A. McKittrick, Dr. R. R. Sayers, Charles Sheard, C. Tasker, R. M. Watt, Jr., and H. E. Ziel. Previous reports on the subject are listed in the bibliography.

The usual conception of air conditioning for human comfort assumes its application for the purpose of providing ideal comfort. A field offering a challenge to the air conditioning engineer is rapidly developing in which air conditioning is applied to alleviate undue discomfort and injury to the health of workers in hot industries. This has sometimes been referred to as air conditioning in industry. In either case, the atmospheric environment is controlled in order to gain the desired effect on the physiological responses including the feeling of comfort or discomfort. In one case the person benefited by air conditioning expects to enjoy the optimum condition for a comfortable existence; while in the other case it is frequently assumed that the best that can be afforded is an environment which will alleviate great distress, and avoid violent physiological derangement which may either cause excessive current suffering or permanent injury to the health of the worker, reduce his output, or cause him to avoid employment at the work in question.

The relations between the physical activity of the person and his rate of heat production on the metabolism on the one hand, and the rate of dissipation of this heat to the atmospheric environment on the other hand have been amply established for a wide range of activities and atmospheric conditions through research by the ASHVE Laboratory in cooperation with the U. S. Bureau of Mines, the U. S. Public Health Service, and others. These studies show that the rate of heat production ranges from 53 cal per square meter per hour when seated at rest to as high as 180 cal per square meter per hour at very hard labor or 118 cal per square meter per hour for labor construed to be as hard as the average worker cares to be engaged in continuously for two 4-hour work periods per day. For a person at rest this heat may be dissipated with ideal comfort of from 66 to 72 deg ET. Perspiration occurs at about 76 deg ET and a rise in body temperature and other physiological derangement begins to occur when the temperature reaches about 82 deg ET. For a person engaged in continuous labor with a metabolic rate of 118 cal per square meter per hour a lower effective temperature is required for ideal comfort, and a rise in body temperature and other physiological derangement occurs at about 75 deg ET. The relation between rise in body temperature and the environmental conditions for a person at light work taken from an earlier laboratory study[1] are shown in Fig. 1. It is obvious from these facts that in

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**Advisor on Physiological Studies at the ASHVE Research Laboratory. Research Fellow, Dept. of Industrial Hygiene, College of Medicine, University of Pittsburgh. MEMBER of ASHVE.
†Research Assistant, ASHVE Research Laboratory.

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, San Francisco, Calif., June, 1941.

[1]Numbers refer to Bibliography.

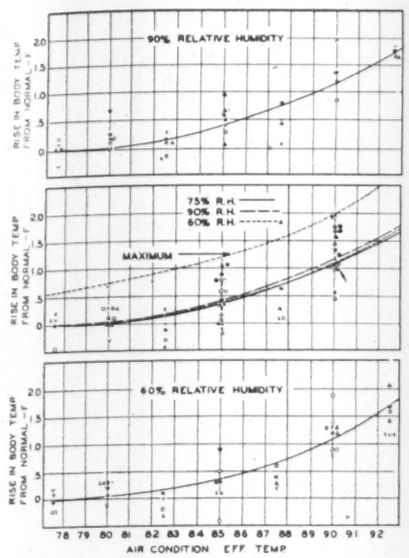
90% RELATIVE HUMIDITY
75% R.H.
90% R.H.
60% R.H.
MAXIMUM
60% RELATIVE HUMIDITY
RISE IN BODY TEMP FROM NORMAL °F
78 79 80 81 82 83 84 85 86 87 88 89 90 91 92
AIR CONDITION EFF TEMP

Fig. 1—Relation between rise in body temperature from normal during 3-hour exposure and the effective temperature of the air condition in the test room for 60, 75 and 90 per cent relative humidities. Test points for individual subjects apply to heavy solid curves

order to avoid violent physiological derangement alleviation is necessary in many industries if men are to work regular shifts. The air conditioning engineer is naturally called upon to supply improved conditions through air conditioning.

Where the occupancy density in a hot industry is high, the most logical, as well as the most feasible, course is to air condition the entire environment so as to supply ambient air in the entire enclosure sufficiently satisfactory to give the desired results. There are many instances, however, where one or at most a small number of men are engaged in a large enclosure in which an industrial process requiring, or resulting in, hot surroundings is located. For such conditions the conditioning of the entire space is highly uneconomical or even impossible from a practical point of view. Here, the application of local cooling to take care of the few occupants without conditioning the entire ambient surroundings may be resorted to by use of one of the following methods: (1) In some instances the worker may be placed in a conditioned booth from which he may observe operations and perform his required tasks. (2) The worker may be supplied with a blast of cool, high velocity air into which he may step, thus providing himself with the degree of cooling required. (3) Cooled air may be blown through a loose-fitting, relatively air-tight suit so as to cool the worker without attempting to cool the surroundings. This paper includes the results of a study of the second and third methods of local cooling applications, the results being limited to atmospheric conditions of approximately 50 per cent relative humidity and either a 2500 fpm blast of cool air from a 4-in. nozzle located close to the worker, or a cool air supply of approximately 20 cfm through a loose-fitting *coverall* suit.

The study was made in the psychrometric rooms of the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in the U. S. Bureau of Mines Building, Pittsburgh, Pa. Two subjects, one referred to as the *cooled worker* and the other as the *control worker*, spent the test period in one of the two psychrometric rooms in which the desired ambient air condition at approximately 50 per cent relative humidity was maintained. The cooled air was provided by a small forge type blower from the second psychrometric room so conditioned as to give cooling air of the desired dry-bulb and effective temperature with approximately 50 per cent relative humidity at the location of the worker.

In the blast study, the cooled worker spent most of his time either standing or sitting at rest in the blast of cool air, orienting himself at will to obtain the most desired effect. However, during approximately 1 min out of each 5-min period he left the cool blast to pull the work machine 10 times. This resulted in his being at ease in the cool blast approximately 80 per cent of the time and at work in the ambient condition the remaining 20 per cent of the time. The control worker divided his time between *at work* and *at ease* in a like manner. However, he was at no time permitted to enter the blast of cool air.

In the ventilated suit study, air of the desired dry-bulb and effective temperatures at approximately 50 per cent relative humidity was continually circulated through the suit of the cooled worker. The temperature of the ventilating air was observed in the 2-in. hose near the end where it was connected to the back of his *coverall* suit. During each 5-min period he was required to pull the work machine (Fig. 2) 10 times, which required about 1 min; the remaining approximately 80 per cent of the time he was allowed to sit or stand at ease. The control worker followed the same procedure with the exception that he was not supplied with cooling air through his suit.

The subjects were healthy young men from 18 to 22 years of age and of average size (approximately 19.5 sq ft body surface area) who were able to satisfactorily pass a physical examination given by a physician with industrial experience and classified by him as being good

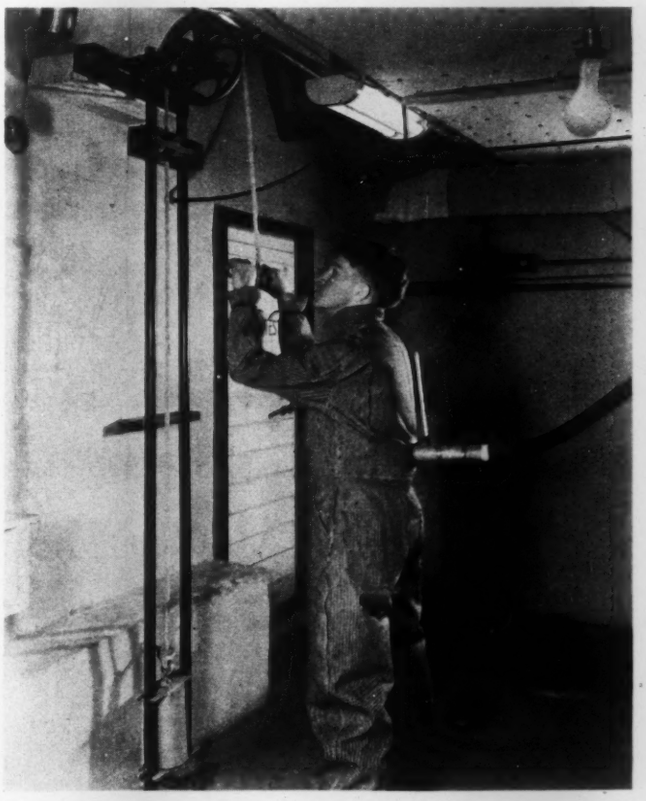

Fig. 2—Worker in ventilated suit operating work machine

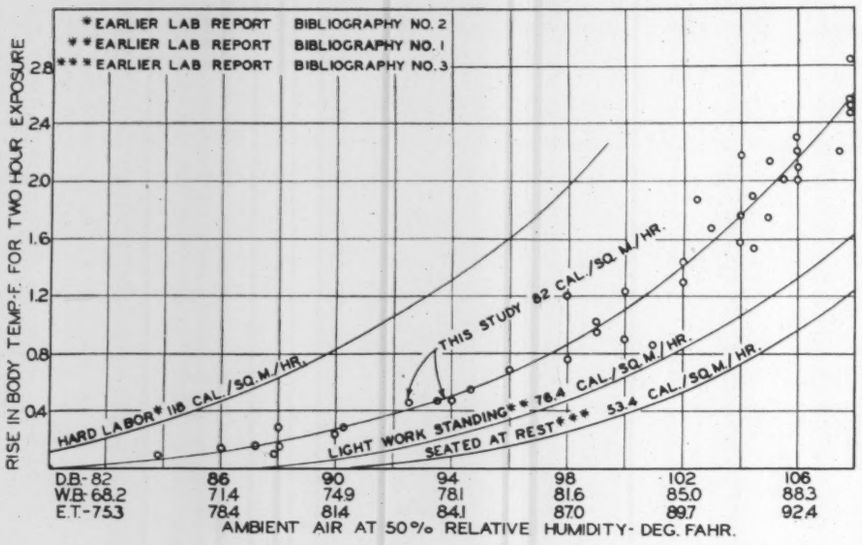
*EARLIER LAB REPORT BIBLIOGRAPHY NO. 2
**EARLIER LAB REPORT BIBLIOGRAPHY NO. 1
***EARLIER LAB REPORT BIBLIOGRAPHY NO. 3
RISE IN BODY TEMP-F. FOR TWO HOUR EXPOSURE
HARD LABOR* 118 CAL./SQ.M./HR.
THIS STUDY 82 CAL./SQ.M./HR.
LIGHT WORK STANDING** 78.4 CAL./SQ.M./HR.
SEATED AT REST*** 53.4 CAL./SQ.M./HR.
D.B.-82 86 90 94 98 102 106
W.B.-68.2 71.4 74.9 78.1 81.6 85.0 88.3
E.T.-75.3 78.4 81.4 84.1 87.0 89.7 92.4
AMBIENT AIR AT 50% RELATIVE HUMIDITY-DEG. FAHR.

Fig. 3—Relation between ambient air and rise in body temperature of control worker. Curves for other activities given for comparison

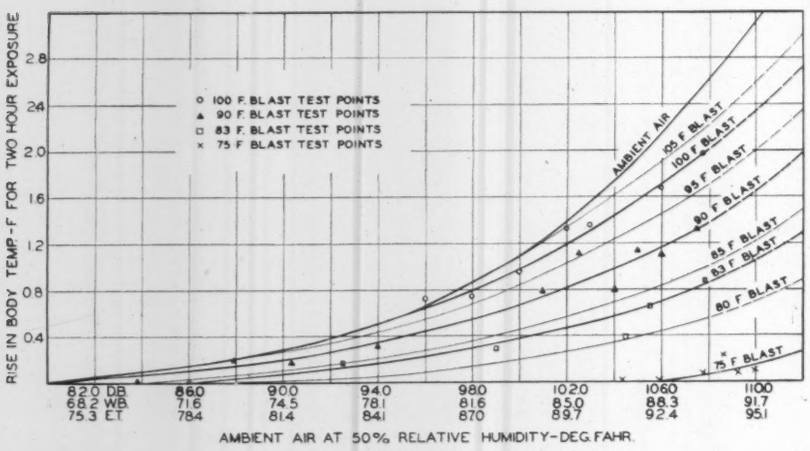
100 F. BLAST TEST POINTS
90 F. BLAST TEST POINTS
83 F. BLAST TEST POINTS
75 F. BLAST TEST POINTS
RISE IN BODY TEMP-F FOR TWO HOUR EXPOSURE
AMBIENT AIR
105 F BLAST
100 F BLAST
95 F BLAST
90 F BLAST
85 F BLAST
83 F BLAST
80 F BLAST
75 F BLAST
82.0 DB 86.0 90.0 94.0 98.0 102.0 106.0 110.0
68.2 WB 71.6 74.5 78.1 81.6 85.0 88.3 91.7
75.3 ET 78.4 81.4 84.1 87.0 89.7 92.4 95.1
AMBIENT AIR AT 50% RELATIVE HUMIDITY-DEG. FAHR.

Fig. 4—Relation between ambient air temperature and rise in body temperature of worker cooled by blast air at 50 per cent relative humidity and various dry-bulb temperatures

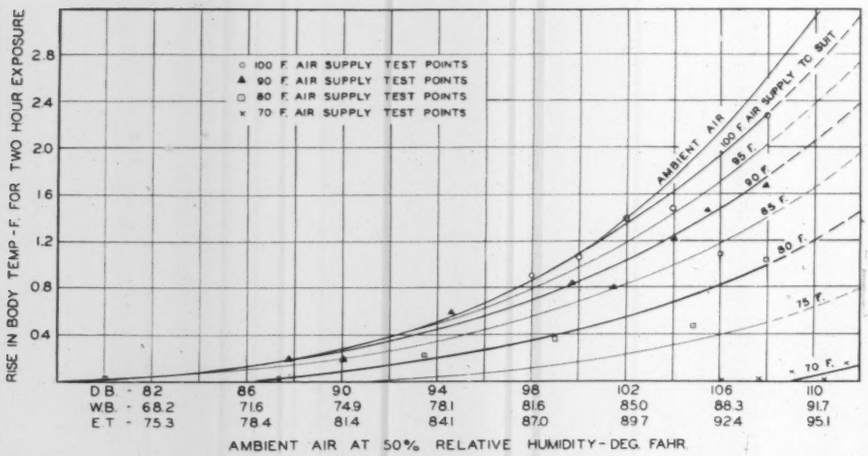
100 F. AIR SUPPLY TEST POINTS
90 F. AIR SUPPLY TEST POINTS
80 F. AIR SUPPLY TEST POINTS
70 F. AIR SUPPLY TEST POINTS
RISE IN BODY TEMP-F. FOR TWO HOUR EXPOSURE
AMBIENT AIR
100 F AIR SUPPLY TO SUIT
95 F.
90 F.
85 F.
80 F.
75 F.
70 F.
D.B. - 82 86 90 94 98 102 106 110
W.B. - 68.2 71.6 74.9 78.1 81.6 85.0 88.3 91.7
E.T. - 75.3 78.4 81.4 84.1 87.0 89.7 92.4 95.1
AMBIENT AIR AT 50% RELATIVE HUMIDITY-DEG. FAHR.

Fig. 5—Relation between the ambient air temperature and rise in body temperature of worker cooled in ventilated suit, with ventilating air at 50 per cent relative humidity and various dry bulb temperatures

risks for workers in hot industries. In all, six different men served as subjects, two in each test. On succeeding days when the same subjects participated they alternated their positions as *cooled* and *controlled* workers. The clothing worn by all subjects in all tests included light weight underwear, low shoes, light weight socks, and coveralls.

The subjects entered the test room after resting 15 min in a relatively comfortable room. Just before entering the test room their body temperature, pulse rate, general feeling of warmth, and other physiological data were recorded. The collection of these data was continued through the period spent in the test room. Attention was particularly paid to the rise in body temperature, increase in pulse rate, degree of perspiration, and the comfort or discomfort experienced by the worker. In a few of the tests the metabolism for this degree of activity was determined and found to be 82 cal per square meter per hour. The subjects continued in the test room for 2 hours unless their body temperature reached 101 deg or their pulse 140 beats per minute, which were accepted as the safe limits beyond which the men should not be exposed. When the reactions of either subject reached either of these limits he was immediately removed from the test which was however continued with the other subject.

Test Results

The rise in body temperature of the control worker, in both the blast and suit tests, was plotted against the dry-bulb temperature of the ambient condition at 50 per cent relative humidity in Fig. 3. The accompanying wet-bulb and effective temperatures of the ambient condition are indicated. For comparison, curves are included showing the relation between conditions of the atmosphere and rise in body temperature found in an earlier study for a person working at relatively hard labor,[2] a person working at very light work[1] moving about on his feet, and for a person seated at rest[3] all for still air conditions.

The relation between the rise in body temperature of the worker cooled by blast air at several dry-bulb temperatures at 50 per cent rel-

ative humidity and the ambient conditions are plotted in Fig. 4. Tests were actually made at, and the test points are plotted for, blast temperatures of 100 deg, 90 deg, 83 deg, and 75 deg dry-bulb. Heavy line curves for these points are drawn. These curves are expanded into a series giving the relationship between temperature rise of the worker and blast temperatures ranging from 100 deg to 75 deg dry-bulb at 50 per cent relative humidity.

The relation between the rise in body temperature of the worker cooled by ventilation of his suit is plotted against the dry-bulb temperature of the ambient condition at 50 per cent relative humidity in Fig. 5. Tests were actually made and the test points are plotted for ventilating air dry-bulb temperatures of 100, 90, 80, and 70 F, and these curves are expanded into a series giving the relation between rise in body temperature and ventilating air temperatures ranging from 100 to 70 deg dry-bulb at 50 per cent relative humidity.

The increase in pulse rate for the control and cooled workers was also observed and studied; however, they show approximately the same relation to rise in body temperature found in earlier studies.[1] Since rise in body temperature has been found to be the best and most consistent physiological change in its correlation with the severity of the hot condition, only the rise in body temperature is here reported.

The control worker perspired profusely in the hot ambient conditions. While in the light work study,[1] Fig. 3, perspiration began to be noticed on the subjects at from 75 to 78 deg ET, the control subjects in this study engaged in the higher work output began to perspire sensibly at from 72 to 75 deg ET. Further, while for the light work studies perspiration did not soak completely through the clothing of the workers until an effective temperature of 88 or 90 deg was reached, this condition was observed for the control worker in this study at about 86 deg ET. The worker cooled by either the blast or the ventilated suit in this study was relatively free from perspiration for all ambient conditions studied, when the cooling air temperature was as low as 80 deg dry-bulb with 50 per cent relative humidity. Until an ambient effective temperature of 95 deg, or a dry-bulb of 110 F at 50 per cent relative humidity was reached, 60 deg dry-bulb cooling air eliminated sensible perspiration entirely.

The data plotted in Figs. 3, 4, and 5 show that for the type of work engaged in by the control worker in this study he began to show a rise in body temperature when the dry-bulb temperature with 50 per cent relative humidity reached 80 F (or 74 deg ET) and he had a temperature rise of 1 deg in 2 hours when the dry-bulb temperature reached 99 deg. With ambient conditions of 80, 88, and 106 there was no rise in the body temperature of the cooled worker in blast temperatures of 90, 80, and 75 F, respectively. With the ventilated suit, slightly lower air temperatures were necessary. A few tests made with 60 deg air show that workers can perform the amount of work employed in this study for dry-bulb temperatures up to at least 112 deg with 50 per cent relative humidity.

Although the study was made with both ambient and cooling air temperatures at 50 per cent relative humidity, earlier work of the Laboratory showing that most physiological reactions, particularly rise in body temperature, are functions of effective temperatures, makes it possible to apply the results to other atmospheric conditions.

The study shows conclusively that workers may be kept free from body temperature rise, excessive increase in pulse rate, and from perspiration either with a blast of cool air or by ventilating their clothing with cool air. It should be emphasized, however, that variations in the manner in which the cooling air is handled or in the clothing worn may materially affect these results. Larger volumes of blast air or ventilating air through the suit would obviously give greater cooling effect and permit the worker to operate at higher ambient temperatures or in the same condition for longer periods of time. Likewise the same volume of air at a lower effective temperature will have the same effect.

The type of clothing worn is important. The cooling air will obviously have greater effect the closer it is kept to the body surface. In very hot conditions, particularly if radiant heat is an important factor, a given volume of cooling air blown through the suit will be more effective with greater insulation outside of the loose-fitting air tight suit; however, heavy clothing is objected to by the worker and should be avoided whenever possible.

Blast cooling air will be more effective the more intimately it comes in contact with the worker. In this connection it should be observed that after leaving the nozzle the cool air jet mixes rapidly with ambient air so that its temperature rises as the distance from the nozzle increases. The nozzle should be located near the worker. In this study it was found most effective to place the nozzle near the shoulder of the standing worker with the axis of the air stream projected downward at an angle so that the cooling air stream enveloped as large a portion of the body surface as possible. The worker was found, in general, to object to the blast of air over his head or face even though the sensation of coolness might be desirable.

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LEE P. HYNES MOVES

Lee P. Hynes, electrical and mechanical engineer, has announced the removal of his office from 240 Cherry St., Philadelphia, to West and Clinton Sts., Camden, N. J. Mr. Hynes has been a member of the ASHVE since 1919, and is also a member of the *AIEE* and *ASME*.

NEW OFFICERS INSTALLED AT NEW HAVEN

May 21, 1941. The first annual meeting of the Connecticut Chapter was held in the Faculty Club, New Haven, with Pres. L. E. Seeley presiding. The meeting, which was preceded by a dinner and get-together, was called to order by the president.

The principal items of business were the reports by the president and treasurer, the admission of new members and a resolution expressing the appreciation of the members to the first secretary-treasurer, E. J. Rodee, for all of the time and thought he devoted to the organization of the chapter. Mr. Rodee has been called to active service in the U. S. Navy and is now stationed at the Fore River Shipbuilding Plant in Quincy, Mass.

New officers for the year were elected as follows:

President—W. K. Simpson
Vice-President—C. K. Lyons
Treasurer—C. G. Chapin
Secretary—L. A. Teasdale
Board of Governors—L. E. Seeley, Stanley Hart and J. R. Smak.

After the business meeting W. Walter Timmis, New York, gave an interesting talk on weather controls, their application to the heating systems of buildings and the probable economies in fuel consumption to be expected.

WESTERN MICHIGAN INSTALLS NEW OFFICERS

May 23, 1941. The annual meeting of the Western Michigan Chapter was held at Green Ridge Country Club, Grand Rapids, and the afternoon was spent at horse-shoes, golf and archery. The Chapter Cup for golf was awarded to B. F. McLouth.

Dinner was served to 80 members and guests, following which the president appointed Messrs. De Roo, Dean and McMaster as tellers and after the ballots were counted the following were declared elected for the 1941-42 season:

President—W. G. Schlichting
Vice-President—F. C. Warren
Secretary—H. J. Metzger
Board of Governors—T. D. Stafford, H. D. Bratt and C. H. Pesterfield.

Suitable ceremonies took place under the direction of Wendell Warnock for initiating the new officers into their respective offices.

H. D. Bratt reported on the condition of C. H. Morton and flowers will again be sent to Mr. Morton.

Past President Stafford expressed his appreciation to all of the various committees and officers who assisted him in making the year a success for the Chapter.

There were several high lights of the evening among which were presenting of a silver platter to Past President Miller for his continued time and effort put forth for the good of the Chapter especially in regard to the furnishing of the past presidents' plaques.

W. W. Bradfield presented Reynolds number medal to S. H. Downs and henceforth Mr. Downs will undoubtedly be known as *Reynolds Number Downs*. D. L. Taze made considerable effort to attend the meeting by coming over from Cleveland and was duly presented with his past president's plaque. After W. G. Schlichting became president he had the honor to present Retiring President Stafford with his past president's plaque.

The auditing committee was then appointed consisting of Messrs. S. W. Todd, Jr., and C. R. McConner.

F. C. Warren, secretary of the Chapter during the past year, expressed his appreciation to all the local members who willingly gave their assistance during the year, and the meeting adjourned at 10:45 p. m.

April 14, 1941. The April meeting of the Western Michigan Chapter was held at the Mechanical Engineering Laboratory at Michigan State College, with the engineering department of the College acting as hosts, and 46 members and guests were in attendance.

The meeting was called to order by W. G. Schlichting in the absence of President Stafford, and the minutes of the previous meeting were read and approved. The nominating committee reported as follows: *President* — W. G. Schlichting; *Vice-President* — F. C. Warren; *Secretary*—H. J. Metzger; *Treasurer*—A. H. Snook; *Board of Governors*—T. D. Stafford, *Chairman*; H. D. Bratt and C. H. Pesterfield. Nominations were called for from the floor, but, none being offered, motion was made and seconded that the nominations be closed, and motion was carried upon vote.

Mr. Warnock, chairman of the entertainment committee, reported on the progress of plans for the annual meeting scheduled for the Green Ridge Country Club. This was followed by the presentation of past president's plaques by O. D. Marshall to S. H. Downs, Prof. L. G. Miller and C. R. McConner.

Professor Reuling introduced the first speaker of the evening, Russell Lloyd, who gave a talk on photoelastic measurements. Professor Miller introduced the next speaker, Mr. Simmons of the Republic of Panama, who gave considerable data on the Republic of Panama as to population, size of cities, etc.

Vice-President Schlichting expressed the Chapter's appreciation to Dean Dirks, Professor Miller and Professor Reuling for their hospitality.

Motion was made, seconded and carried that flowers be sent to Pres. T. D. Stafford, which was followed by adjournment.

JAMES ADDRESSES WASHINGTON, D. C. CHAPTER

May 14, 1941. The regular monthly meeting of the Washington, D. C., Chapter was held at the Dodge Hotel, and before introducing the guest speaker the following officers were elected and inducted into office:

President—F. A. Leser
Vice-President—R. S. Dill
Secretary—F. M. Thuney.
Treasurer—K. F. Baldwin, Jr.
Board of Governors—F. E. Spurney, Rutcher Skagerberg and A. E. Stack.

General Wilmot A. Danielson, U.S.A., who was visiting from the Panama Canal Zone, inducted the newly elected officers, with the able assistance of Commander T. H. Urdahl, U.S.N.

A. E. Stacey, Jr., Chairman of the Society's Committee on Research, introduced John James, New York, Technical Secretary of the ASHVE, who delivered an enlightening talk on Research and Development in Heating, Ventilating and Air Conditioning. Mr. James' address was illustrated with slides and was well received by his audience, which numbered 138 members and

guests. The entire class of National Defense Training Course in Air Conditioning at George Washington University was among those present. Mr. James' talk was a subject of great interest, as evidenced by the discussion which took place before the meeting adjourned.

ST. LOUIS ELECTS OFFICERS

May 6, 1941. The regular meeting of the St. Louis Chapter was held at the Kingsway Hotel with 29 in attendance.

C. F. Boester read the minutes of the previous meeting which were approved as read.

Pres. C. E. Hartwein read the report of the nominating committee and the following were declared elected to serve for the coming year:

President—D. J. Fagin
First Vice-President—M. F. Carlock
Second Vice-President—C. F. Boester
Secretary—J. H. Carter
Treasurer—W. J. Oonk
Board of Governors—C. E. Hartwein, B. C. Simons, L. J. DuBois and J. S. Rosebrough.

Retiring President Hartwein then turned the meeting over to L. Walter Moon, who installed the new Chapter officers.

After a brief recess, the meeting was again called to order and Mr. Carter introduced Messrs. Sampson and Norris who discussed the paper by W. L. Fleisher, Sr., and W. L. Fleisher, Jr.: *Comfort and Health and Temperature a Mathematical Solution.* This paper was presented in a very able manner and a spirited discussion followed. Chairman Carter then introduced L. J. DuBois, York Ice Machinery Corp., who spoke on the developments in refrigeration for air conditioning. This was an interesting discussion of the history and progress of design and application of refrigerating equipment to air conditioning.

April 1, 1941. The regular Chapter meeting was held at the Kingsway Hotel.

In the absence of the president and vice-president, the meeting was called to order by M. F. Carlock, Second Vice-President. Minutes of the March meeting were read and approved.

Chairman Tenkonohy of the Nominating Committee then announced that the unanimous selection of the Committee for officers to be elected at the annual meeting were the following: *President*—D. J. Fagin; *First Vice-President*—M. F. Carlock; *Second Vice-President*—C. F. Boester, Jr.; *Secretary*—J. H. Carter; *Treasurer*—W. J. Oonk; *Board of Governors*—C. E. Hartwein, B. C. Simons, L. J. Du Bois and J. Rosebrough.

Mr. Tenkonohy pointed out that it was the recommendation of the Committee in this slate that the secretary and treasurer serve only one year instead of two years, as previously. He also called the Chapter's attention to the fact that the Committee had urged G. B. Rodenheiser to accept an office on the slate in view of his long service in the interest of the Chapter. As on previous occasions, Mr. Rodenheiser declined due to press of other matters, but the Committee felt that some definite form of appreciation should be shown to Mr. Rodenheiser.

After a short recess, Mr. Carter introduced the speaker of the evening, J. R. Vernon, Johnson Service Co., Chicago, who spoke on *Automatic Temperature Control for Air Conditioning Systems*, which was very ably presented, giving those present a broad view of the temperature control industry and its facilities for serving the air conditioning and heating industries.

WESTERN NEW YORK HOLDS ELECTION

May 19, 1941. The Western New York Chapter held its May meeting at the University Club, Buffalo, and, following the reading of the minutes of the April meeting, Pres. C. A. Gifford appointed Messrs. Adema, Davis and Spencer as tellers to count the ballots cast in the election of 1941-42 officers.

B. C. Candee submitted the report of the treasurer, which was read, approved and recorded with a rising vote of thanks to Mr. Candee.

The tellers reported that the following officers had been elected:

President—W. R. Heath
First Vice-President—H. C. Schafer
Second Vice-President—S. M. Quackenbush
Treasurer—B. C. Candee
Secretary—Herman Seelbach, Jr.
Board of Governors—M. C. Beman, Joseph Davis, Roswell Farnham and Cecil Farrar.

Mr. Farrar reported on the Toronto meeting and outlined the future program toward bringing the 1944 Society meeting and heating and ventilating show to Buffalo.

President Gifford, before relinquishing the gavel to the new president, thanked the members and officers for their fine cooperation and then called on Mr. Davis to install the new officers.

Mr. Heath then took the chair as president, and after some discussion of a summer golf meeting, President Heath named Messrs. Strouse, Davis and Mahoney to serve as a committee to make arrangements for such a meeting.

The meeting was then turned over to the Board of Publicity of the City of Buffalo, who showed their motion picture entitled *Buffalo, the City of Good Neighbors*, which was viewed with great interest by both members and guests.

OFFICERS INSTALLED AT PHILADELPHIA

May 8, 1941. The regular meeting of the Philadelphia Chapter of the Society was held at the Bala Golf Club, Bala, Pa.

After an afternoon of golf and a general get-together, dinner was served, and at 8:55 p.m. Pres. C. B. Eastman called the meeting to order. By motion, duly seconded, the reading of the minutes and the treasurer's report were omitted.

President Eastman turned the meeting over to H. Berkley Hedges, the newly elected president, who thanked the members for the confidence placed in him, and pledged his best efforts for the good of the Chapter. Mr. Hedges then asked the support of each member in making the coming year a great success.

The officers elected at this meeting to serve the Chapter for the coming year are as follows:

President—H. Berkley Hedges
First Vice-President—H. H. Mather
Second Vice-President—Edwin Elliot
Treasurer—A. C. Caldwell
Secretary—R. D. Touton
Board of Governors — C. B. Eastman and L. E. Moody.

President Hedges, after introducing the newly elected officers, turned the meeting over to W. P. Culbert, chairman of the golf committee, to take charge of awarding the golf awards, which had been donated by various members of the Society. The meeting adjourned at 9:15 p.m.

ONTARIO ELECTS OFFICERS

May 5, 1941. The annual May meeting of the Ontario Chapter was held at the Royal York Hotel and 39 members and guests were present.

Following dinner, Pres. J. W. O'Neill called upon R. W. Bayles, who gave a report on the financial status. He then read the following resolution which the Board of Governors asked the Chapter to vote on:

That the Board of Governors be authorized to invest surplus funds in the new War Loan to an amount not exceeding that necessary to leave at least $300.00 in the treasury.

The resolution was duly moved and seconded and unanimously adopted by the Chapter.

W. R. Blackhall made a report on the University of Toronto Award, and stated that out of 34 students in the mechanical engineering class of the University, six students wrote on heating, ventilating and air conditioning. He announced that the award for the current year was made to D. L. Angus, who wrote on the heating of buildings by means of unit heaters.

J. P. Fitzsimons made a report on the program committee and reviewed the meetings of the season from October 1940 to date. S. A. Jennings reported on the activities of the greeters' committee for the past year, which was followed by a report of the attendance committee by C. D. Waldon. Mr. Waldon announced that the average attendance for the past year was approximately 64.

W. C. Kelly made a report on the smoke abatement activities of the City of Toronto. He further stated that there had been several meetings at which various recommendations had been heard and received. However, he stated that there would be no definite decision on this matter until next fall.

President O'Neill announced the names of the candidates for officers for the Ontario Chapter for the coming season, and moved by M. W. Shears and seconded by H. B. Jenney, the following were unanimously elected:

President—C. Tasker
Vice-President—D. O. Price
Secretary-Treasurer—H. R. Roth
Board of Governors—W. C. Kelly, E. R. Gauley and A. S. Morgan.

President O'Neill then turned the chair over to Mr. Tasker, the incoming president. H. D. Henion, on behalf of the Chapter, presented retiring President O'Neill with a certificate expressing the esteem of the Chapter for a very successful season.

April 21, 1941. The regular monthly meeting of the Ontario Chapter was postponed so that it would coincide with the Society's Council meeting held in Toronto. Dinner was served at 6:30 with 132 members and guests in attendance, and at the conclusion, E. Holt Gurney extended a welcome to the visiting members of the Council. Pres. W. L. Fleisher responded to Mr. Gurney's welcoming address.

President O'Neill then asked Prof. E. A. Allcut to introduce Prof. E. O. Eastwood, Seattle, Wash., first vice-president of the ASHVE, who spoke on the developments in the aircraft industry, and particularly the developments of heating, ventilating and air conditioning in this industry. Professor Eastwood illustrated his talk with slides which were both interesting and instructive.

C. Tasker then introduced Prof. A. P. Kratz, University of Illinois, Urbana, and a member of the Council. Professor Kratz gave an interesting talk on warm air heating in homes, which he illustrated with slides.

E. R. Gauley then extended a vote of thanks to the speakers and to the Royal Norwegian Air Force, who were hosts to the Chapter in the afternoon when they visited Little Norway.

The Chapter arranged for a luncheon in honor of the Society Officers and Council at the Granite Club on April 21, and 75 were present to hear Pres. W. L. Fleisher give a brief talk on the Aims, Activities and the Future of the Society.

DELTA ANNOUNCES NEW OFFICERS

May 1, 1941. The Delta Chapter at its monthly meeting held its second *Information, Please* program. The quiz was on temperature controls, and according to its report, this quiz was such a success that another *Information, Please* program was scheduled for the next meeting on the topic of air distribution.

According to the secretary's report the Delta Chapter highly recommends the *Information, Please* program to any of the Chapters. He pointed out that at this meeting practically every member present is brought into the discussion, thus giving the members, who are not so well versed on the particular subject, an opportunity to hear a discussion in non-technical language thereby giving them an insight into subject matter planned for the quiz.

The secretary, in his report, explained that cards are mailed to the members and returned unsigned, with the questions to be submitted, and from then on the program is conducted very much in the same manner as the various radio quiz programs; that is, three members

DELTA CHAPTER OFFICERS

C. B. Gamble
President

G. C. Kerr
Vice-President

O. E. Gammill, Jr.
Treasurer

F. G. Burns
Secretary

are named judges and four comprise the contestants. The President acts as quizzer.

During the meeting, as reported by Secretary Burns, the chairman of the nominating committee, L. V. Cressy, announced the following as the officers elected to serve the Delta Chapter for the coming year:

President—C. B. Gamble
Vice-President—G. C. Kerr
Treasurer—O. E. Gammill, Jr.
Secretary—F. G. Burns
Board of Governors—L. S. Weil, J. J. Friedler, Jr., and G. E. May.

ICE ACCUMULATORS TOPIC AT ATLANTA

May 6, 1941. The regular meeting of the Atlanta Chapter was held at the Atlanta Biltmore Hotel and was called to order by Pres. S. W. Boyd, with 17 members present.

The minutes of the previous meeting were read and approved, followed by a report by W. J. McKinney who reported that the Chapter's finances were in satisfactory condition.

President Boyd, in the absence of T. T. Tucker, advised the Chapter that the annual engineer's outing at East Lake, of which Mr. Tucker is chairman, would be held on May 30, and all were requested to give him their support in order to make this function a success.

The secretary then read several letters recently received from the headquarters in New York, with reference to the Semi-Annual Meeting in San Francisco, June 1941. Any member planning to attend this meeting was requested to get in touch with the secretary and the information on hand would be made available to him.

President Boyd then introduced the guest speaker, C. J. Otterholm, Minneapolis, Minn., who gave an interesting, well-founded, and well prepared talk on the use of ice accumulators for storage air conditioning systems. The speaker pointed out that his company had recently completed some performance tests on these units and that complete design and performance data would soon be available on this equipment. It will therefore be possible for an engineer to accurately size this equipment for a job just as is now being done with compressors.

The speaker also pointed out that many power companies are making reduced rates available for off peak periods which make it possible to show a saving of as much as 50 per cent on the power bill when using ice accumulators. The speaker suggested that it would be advantageous to the air conditioning engineers in this territory to present this thought to the local power company.

Following his talk, the speaker answered numerous questions and a very lively discussion resulted.

President Boyd then thanked Mr. Otterholm for his interesting talk and also for paying the Atlanta Chapter this visit even though it involved considerable traveling on his part.

MANITOBA NOMINATES OFFICERS

April 24, 1941. The regular monthly meeting of the Manitoba Chapter was held at the Fort Garry Hotel with Pres. P. L. Charles presiding. The minutes of the previous meeting were accepted as read, and President Charles read the report of the nominating committee, who had submitted the following names for officers for the coming season: *President*—R. L. Kent; *Vice-President*—Ivan McDonald; *Secretary-Treasurer*—F. L. Chester.

In discussing the formation of a Board of Governors, it was moved by G. C. Davis that this Board should consist of past presidents of the Chapter, in order of their retirement. This movement was approved by the members present, and President Charles requested that a notice be sent to chairmen of the various committees to submit their reports at the final meeting of the year in May. It was suggested by Mr. Davis that recognition of the services of the retiring secretary-treasurer should be suitably expressed, and this was unanimously approved by all members.

PROFESSOR EASTWOOD ADDRESSES WISCONSIN

April 22, 1941. The April meeting of the Wisconsin Chapter was held at the City Club with 55 members and guests in attendance. After dinner Pres. A. S. Krenz announced the forthcoming annual Engineering Conference to be held at the Pfister Hotel, Milwaukee, and invited all members to attend.

C. H. Randolph then called attention to the May meeting to be held at Madison, and stated that an excellent program was being prepared with the help of the Madison members. President Krenz then turned the meeting over to Mr. Randolph, chairman of the program committee, who introduced the evening's speaker, Prof. E. O. Eastwood, Seattle, Wash., first vice-president of the ASHVE. Professor Eastwood gave a very interesting and timely talk on *An Engineer's View of Defense*. He covered the heating,

DELTA BOARD OF GOVERNORS

L. S. Weil

J. J. Friedler, Jr.

G. E. May

ventilating and air conditioning of airplanes and also gave some very interesting data on wind tunnels. The lecture by Professor Eastwood was illustrated with slides and was received with a great deal of interest. Upon the completion of his talk, he threw the meeting open to questions, and responded cheerfully to all. In appreciation of his talk the members and guests extended a rising vote of thanks, followed by adjournment.

WARM AIR HEATING DISCUSSED AT NORTHERN OHIO

April 22, 1941. The April meeting of the Northern Ohio Chapter of the Society was held at Hotel Carter, with an attendance of 38 members and guests.

The minutes of the two previous meetings were read by the secretary and approved as read. Pres. C. A. McKeeman read a communication from the Engineering Society about their new building on East 19th St., and then called upon the secretary to report on a meeting, which he had attended, at which there was a discussion of arrangements whereby the Society could hold their meetings at the Engineering Society Building each month. The president asked the program committee to investigate the situation further and if possible arrange for a trial meeting in May.

President McKeeman then announced the appointment of the nominating committee as follows: E. W. Gray, *Chairman;* J. E. Wilhelm and W. R. Rhoton. D. L. Taze of the program committee then asked for comments on a suggested quiz program for the next meeting, which suggestion was not favorably received and the matter was dropped. The meeting was then turned over to the program committee, and the guest speaker, Prof. A. P. Kratz, University of Illinois, was introduced. Professor Kratz gave an interesting review of the work at the Research Residence on warm air heating, and a lively discussion followed, after which the chapter extended a rising vote of thanks to Professor Kratz for his splendid address.

IOWA HOST TO MINNESOTA MEMBER

April 15, 1941. The April meeting was held at the Memorial Union, State University of Iowa, Iowa City, with Pres. R. A. Norman presiding and an attendance of 22 members and guests.

A. L. Walters gave the report of the Nominating Committee duly appointed by the President, recommending the following officers for balloting: *President*—Perry LaRue; *Vice-President*—F. E. Triggs; *Secretary-Treasurer*—C. W. Helstrom; *Board of Governors*—B. E. Landes and M. L. Todd.

Prof. H. O. Croft was then introduced and a vote of thanks was extended him for the splendid way arrangements had been made for this meeting. He in turn introduced F. M. Dawson, Dean of College of Engineering, State University of Iowa. Dean Dawson welcomed the group, and pointed out the importance of codes for municipalities, stressing his feelings that the basic facts behind codes should be investigated and understood. It was his opinion that a group like this should interest itself in codes and bring to their formation and enforcement a high level of engineering intelligence.

President Norman then pointed out that there seemed to be a very decided lack of interest in codes throughout the state and that most municipalities did not have codes or adequate codes. He felt that this was a real field for the ASHVE and that if the Society took an active interest in codes it would probably create more interest on the part of prospective members.

The speaker of the evening, Neil D. Adams, in charge of the Franklin Heating Station, Rochester, Minn., was then introduced. Mr. Adams gave an interesting presentation including a great deal of information about the Mayo Clinic, the history of the Mayo family and the formation of the Clinic and about the Franklin Heating Station, which supplies steam, electricity and hot water to the Clinic group. This most interesting talk was illustrated with slides and was followed by a question and answer period. Among other things, Mr. Adams, who is Chairman of the Society's Membership Committee and has been interested in the Society for a long period of years, pointed out that meetings around the state were a good thing. He suggested that the group of chapters in this northwestern area might very profitably hold a joint meeting or a get-together.

It was the consensus of the group that this was an excellent meeting.

INDUSTRIAL VENTILATION TOPIC AT MASSACHUSETTS

April 15, 1941. The April meeting of the Massachusetts Chapter was held at the Massachusetts Institute of Technology, Cambridge, and 20 members gathered for dinner in Walker Memorial Hall, preceding the meeting, at which A. C. Bartlett presided in the absence of Prof. C. P. Yaglou.

W. C. L. Hemeon of the Massachusetts State Department of Labor and Industries, discussed the subject of industrial ventilation. He explained how, in many cases of ventilation in industries, dilution was the prime problem and many examples were cited, such as lead, gasoline, benzol, etc. Slides of many types of hoods were shown to illus-

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trate how this could be done without causing too much inconvenience or consuming too much space. In this type of work, he pointed out, hood face velocities of from 60 fpm to 200 fpm are necessary, depending upon the type of fume and the distance from the hood to the work.

Mr. Hemeon stated that for some time the *American Standards Association* has been working on a code of toxic limits that will help industry in safeguarding its employees.

Before the meeting was adjourned the new members on the Board of Governors were announced as follows: R. L. Lincoln, R. T. Kern and E. G. Carrier.

PROFESSOR EASTWOOD VISITS MICHIGAN

April 14, 1941. The regular April Meeting of the Michigan Chapter was held at Huyler's L'Aiglon in the Fisher Building, Detroit. There were 48 members and guests present for the meeting, which was called to order at 7:30 p. m. by Pres. G. H. Tuttle.

Reports of the secretary and treasurer were read and adopted; and Vice-Pres. J. S. Kilner discussed the annual golf meeting.

President Tuttle then introduced Prof. E. O. Eastwood, Seattle, Wash., First Vice-President of the Society, who spoke on *An Engineer's View of Defense*, which was followed by a lively question and answer period.

BOILER AND RADIATOR RESEARCH DISCUSSED AT ILLINOIS CHAPTER

April 14, 1941. The regular April meeting of the Illinois Chapter was held at the Merchants and Manufacturers Club, and 95 members and guests attended, which was the largest gathering of the season. A special invitation had been extended by the Chapter to the members of the *Chicago Master Steamfitters Association* and many of them were in attendance.

After the regular order of business Pres. V. L. Sherman called on H. M. Hart to introduce the speakers of the evening whose subjects were I-B-R Research in steam and hot water heating, and the I-B-R Research Home. Mr. Hart first presented Prof. M. K. Fahnestock, Research Associate Professor of mechanical engineering, University of Illinois, who reviewed the activities of the *Institute of Boiler and Radiator Manufacturers* to further the advancement of steam and hot water heating.

The successful formulation and the gaining of acceptance by boiler manufacturers of the I-B-R Testing and Rating Code for Low Pressure Heating Boilers was mentioned as probably the most important technical accomplishment of the Institute to date. It was pointed out that a majority of the manufacturers of cast-iron boilers were endorsing and using the code, and the *Heating, Piping and Air Conditioning Contractors National Association* was including I-B-R ratings in its publication *Net Load Recommendations for Heating Boilers*. The second major technical activity of the Institute was given as the research program, dedicated to the economical and technical improvement of steam and hot water heating for residences.

Mr. Hart then presented W. S. Harris, Research Associate Mechanical Engineer at the University of Illinois, whose talk covered the I-B-R Research Home. With the aid of slides, he described the one-pipe forced circulation hot-water heating system which has been in operation since January 1, 1941, when the tests were started.

F. W. HERENDEEN DIES

Frederick W. Herendeen, Geneva, N. Y., died June 19 at the Clifton Springs Sanatorium, following an operation, at the age of 72. He was born in Geneva, August 25, 1869, and attended the Geneva High School and the Moses Brown School, Providence, R. I. He was graduated from Hobart College in 1892, becoming one of its trustees.

From 1889 to 1912 he held various positions with The Herendeen Mfg. Co., Geneva, becoming vice-president and general manager of sales. In 1912 he became assistant general manager of sales, and later vice-president and a member of the executive board of the U. S. Radiator Corp., Detroit.

In 1915 he became secretary of *The Institute of Boiler and Radiator Manufacturers*, from which he had retired recently. He was a former secretary of the *Gas Heating Boiler and Furnace Association*, and a former President of the *New York League of Savings and Loan Associations*.

He joined the ASHVE in 1919, retaining his membership for over 17 years, and at that time had more than 25 years experience in designing boilers and an intimate first hand knowledge of low pressure heating. Besides being a member of the Society he was also a member of the *ASME*, the *AGA*, and was chairman of the Hobart College Centennial Fund campaign. He was also a trustee of William Smith College.

The Officers and Council of the Society extend their sincerest sympathy to his widow, Mrs. Maude I. Herendeen, who survives.

GOLDEN GATE CHAPTER MEMBER DIES

George Gibson Harrison, a member of the Golden Gate Chapter of the Society since 1937, died at his home in Berkeley, Calif., June 4, 1941. He was born at Copperhill, Tenn., October 7, 1899, and after his graduation from high school completed a course in mechanical engineering at California Polytechnic School, graduating in 1919.

Mr. Harrison entered the employ of the S. T. Johnson Co. in October, 1923, and served continuously as Chief Engineer up to a few weeks before his death.

Since joining the Society, Mr. Harrison has been one of its active members on the Pacific Coast, taking an interest in the affairs of the local chapter, serving on the Board of Governors of the Golden Gate Chapter, as well as on the Reception Committee for the Semi-Annual Meeting of the Society in San Francisco, June, 1941.

Mr. Harrison will be missed greatly by his many friends and associates in the Society, and especially at the monthly chapter meetings, which he attended faithfully.

The Officers and Council of the ASHVE extend their sincere sympathy to his father and mother, Mr. and Mrs. R. F. Harrison, and to his sister, Mrs. Grace Smith, all of Berkeley, Calif., who survive.